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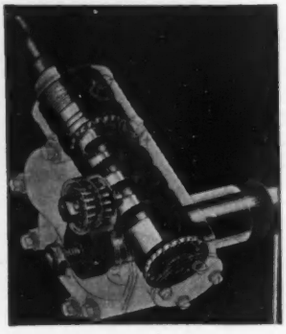
Ease of Wheel-Turn



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. . . Diesels on Canadian Roads . . .

By H. L. Wittek
Consulting Engineer, Toronto, Canada

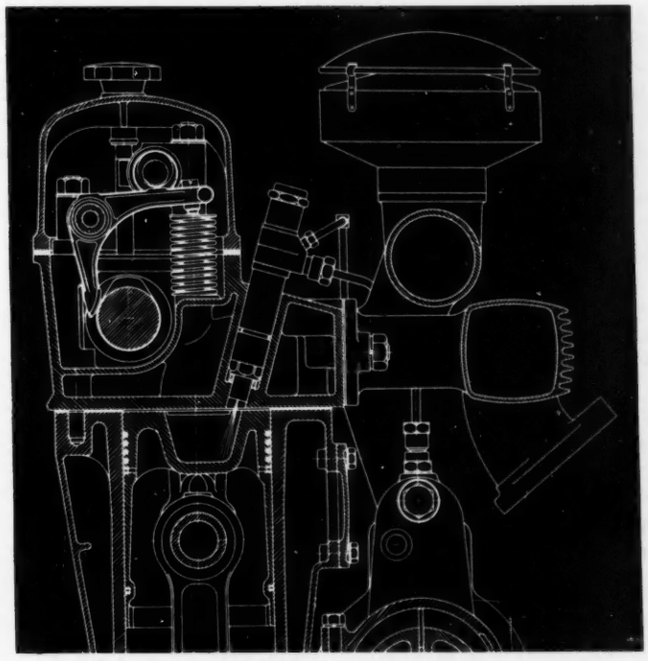


Fig. 1—Combustion Chamber of 496 Cu. In. Piston Displacement Leyland Oil Engine

CANADIAN operators of commercial vehicles did not accept the oil engine as readily and as quickly as did those in most other British territories, in spite of the fact that one of the earliest designs for a high-speed Diesel engine was developed in Montreal. This engine, even in the early experimental stages, had shown considerable promise as a future automotive prime mover. Its potentialities, however, were not appreciated on this continent, and it was left to an English manufacturer to develop from this design the first all-British automotive oil engine. The original combustion chamber design is still in use, although, of course, it has been considerably refined and improved in detail. Its modern form, shown in Fig. 1, is the combustion chamber of the Leyland engine.

When the first oil-engined vehicle was put to work on English roads in 1929, and British and Canadian transport journals reported impressive operating economies, considerable interest in this new development was shown immediately in Canada. As a result of the trade slump, however, Canadian operators were unable to purchase the expensive new equipment, and therefore its application was delayed for a time. Freight shipments were decreasing rapidly and most operators were over-equipped. The fleets consisted mostly of high capacity units which were too large to earn their keep with the small amount of freight being moved. New purchases consisted, therefore, of units of smaller capacity than the Diesel vehicles available at that time. Even when, occasionally, a large unit was purchased, of a size suitable for an oil engine, first cost was still the deciding factor and the gasoline vehicle was chosen.

With the improvement of trade conditions in 1933 and a simultaneous reduction in the price of oil-engined vehicles, road carriers again began to consider seriously the possibilities of the fuel-oil engine. There remained, however, a number of obstacles to be overcome before the use of compression ignition vehicles could become general. Foremost among these was the fact that the few concerns offering oil engines for sale in Canada were opposed by several Canadian truck manufacturers producing gasoline vehicles exclusively, and, of course, the latter had to protect their own interests by advancing all available arguments against the new type. Many of these were perfectly reasonable and included the following:

(*a*) Usable fuel was not available generally and a really suitable fuel could hardly be obtained.

(*b*) At that time no tax was imposed upon Diesel fuels and there was considerable speculation as to what the taxation might be when oil engines became popular. It was considered that a tax might be imposed at such a rate that, with reference to ton miles, it would approximate the gasoline tax, giving a tax rate per gallon of more than twice that imposed on gasoline. Naturally, the operators were unwilling to make a move before this matter was settled.

(*c*) No data were available as to the behavior of oil engines in extremely cold weather.

(*d*) The scarcity of mechanics accustomed to this type of engine might seriously affect the cost of maintenance and service.

These arguments were supported by accounts of the unsatisfactory performance of one or two conversions of gasoline trucks to oil-engine drive, and it was inferred that to purchase an oil engine of any make would necessarily be inviting the same kind of trouble. As this question of converting existing chassis is one that concerns all proprietary engine makers, some of these experiences will be described later.

Nevertheless, a few operators who had observed the development in Europe, decided that the operating economies obtained there were such as to warrant further experiments in spite of the expenditure entailed in initial cost, as well as in

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 19, 1935.]

obtaining the necessary operating experience. It is gratifying that all vehicles purchased within the last two years have rendered such service that their owners have begun to convert further units of their fleets, and that, when new purchases are necessary, oil-engined vehicles are being bought almost exclusively by these operators.

The question of taxation has been settled, the tax rate for engine fuel oil having been definitely established at the same amount per gallon as that for gasoline. The use of oil-engined vehicles has already spread from owners of large fleets to those of one or two vehicles. We are, of course, still lagging far behind developments in the European countries, one example of which is shown in Fig. 2, but the increase in Diesel powered vehicles has recently been so rapid that the general trend in this direction is unmistakable. This tendency has been impeded somewhat during the last few months by the difficulty of obtaining delivery from British manufacturers in less than two or three months, owing to their large commitments. That purchasers are willing to put up with this delay, in spite of the fact that they could obtain immediate delivery on any gasoline vehicle, proves how completely they are convinced.

Canadian customs tariffs have undergone rather frequent changes during the last few years. This caused uncertainty which tended to retard the wider application of British Diesels, as their makers, not knowing what tariff changes might take effect at any moment, were unwilling to spend money in building up new sales organizations. In April of this year, however, an important change was made in the tariff and British-made Diesel engines for buses and trucks are now admitted duty free, thus reducing the laid down cost of the engines by some 15 per cent. As an immediate effect of this regulation Montreal Tramways, which has been operating a few Diesel buses for a considerable length of time, has converted most of its A.E.C. and Leyland coaches, and some other operators of large fleets will follow shortly the same course.

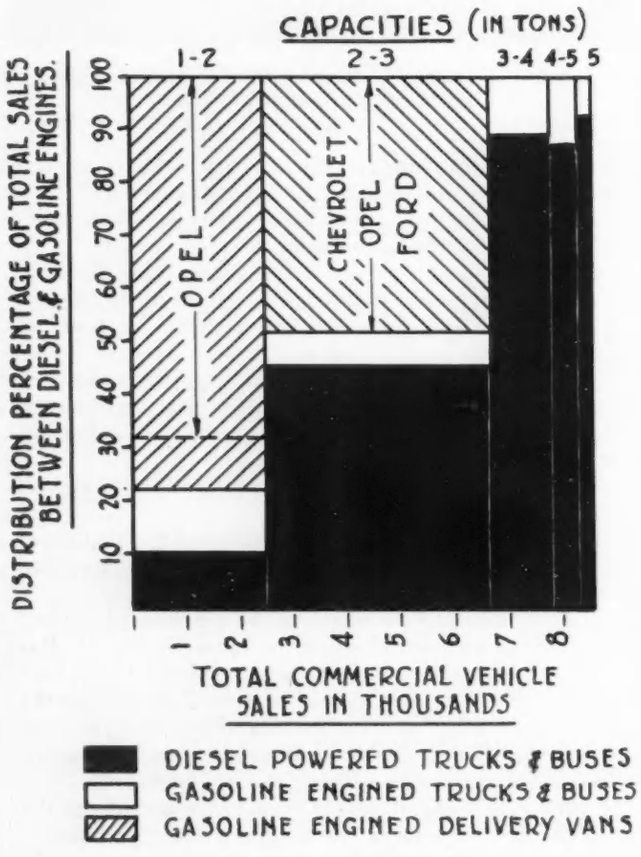

CAPACITIES (IN TONS)
1-2
2-3
3-4 4-5 5
DISTRIBUTION PERCENTAGE OF TOTAL SALES BETWEEN DIESEL & GASOLINE ENGINES.
10 20 30 40 50 60 70 80 90 100
OPEL
CHEVROLET
OPEL
FORD
1 2 3 4 5 6 7 8
TOTAL COMMERCIAL VEHICLE SALES IN THOUSANDS
DIESEL POWERED TRUCKS & BUSES
GASOLINE ENGINED TRUCKS & BUSES
GASOLINE ENGINED DELIVERY VANS

Fig. 2—Commercial Vehicle Sales in Germany, July to December, 1934

The operator is interested only in one problem—how to move his freight, without violating the law, at the least overall expense. As most highway regulations limit gross vehicle-weights as well as overall vehicle dimensions, the problem becomes one of employing the lightest vehicle able to carry the required load, which will entail the lowest cost per mile. The items entering into this calculation are: carrying capacity, first cost, useful life, maintenance cost and direct operating expenses. Engine weight and size are thus of considerable importance and all efforts to reduce them without impairing reliability, first cost and maintenance cost, are desirable. In view of this fact it is surprising that some engine makers use higher weight and greater overall engine length as sales arguments. Experience proves that a shorter engine need not necessarily suffer from weak bearings, whereas some engines with apparently ample bearing areas are still subject to this defect. It appears, therefore, that bearing area alone is by no means a criterion for cost of upkeep. In this connection combustion control, heat transfer from the bearings and bearing materials are at least as important as mere size.

At present, the fixed charges on the Diesel vehicle are considerably higher than those on the gasoline type. Present European oil engines cost about 60 per cent more per hp. and about 16 per cent more per unit of engine weight than gasoline engines of the same power. This results in a price increase for the complete chassis of about 15 per cent. It is, however, to be expected that with increased production the price of oil engines will be reduced, as was the case with gasoline engines. It is also reasonable to assume that the largest factor, at present, in the higher oil engine cost, namely, the fuel injection equipment, will be reduced in due course, thus following the precedent of carburetor and magneto which originally formed a much higher percentage of the total engine cost than they do today.

New and improved designs, together with additional experience, will probably eliminate or greatly reduce the amount of special and expensive materials which are now partly responsible for the greater cost of compression-ignition engines. This trend is very noticeable in engines which have been in production for some time, as in the case of the Leyland engine, where the Duralumin connecting rods, originally employed to overcome big-end difficulties, have been replaced by normal forged connecting rods, using aluminum bearing-shells. The practice of employing a hardened crankshaft, which was necessarily developed in conjunction with the Duralumin rods operating directly on the journals, has been found so advantageous that it is now being used in all engine types of this maker. Several manufacturers producing both types of engine have adopted the practice of designing engines primarily for fuel-oil operation and adapting them for gasoline fuel by changing pistons, cylinders and exhaust valves. In the gasoline engine these valves have to withstand higher temperatures on account of the lower expansion ratio.

As far as maintenance is concerned the experience of Canadian operators has been very satisfactory. A 100 hp. Gardner engine (Fig 3), installed in a G.M.C. Model T-60

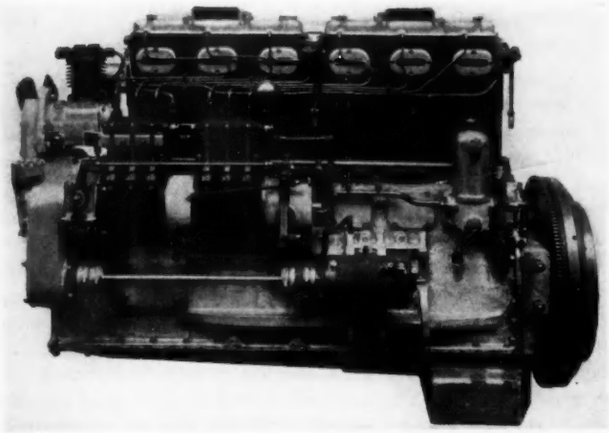


Fig. 3—Gardner 6-LW Automotive Oil Engine. 4¼ in. by 6 in. 102 b.hp. at 1700 r.p.m.

chassis (Fig. 4), has done well over 100,000 miles, operating in temperatures ranging from 40 deg. fahr. below zero to 90 deg. fahr. in the shade. This engine has not yet had a major overhaul, no bearings have been renewed, the cylinders have not been re-ground; the only maintenance work done was one valve grinding, the replacing of a cylinder head gasket and cleaning of sprayers. The engine is hauling a gross load of 55,000 lb. at an average speed of 20 to 22 m.p.h. over a 240-mile run, with some hilly sections. Fuel consumption with this load is between 7.5 and 8.3 miles per gallon, giving the astonishing figure of 206 to 228 gross ton miles per gallon. It should be noted that this performance is obtained in steady normal operation of a properly balanced unit, not by any of the artificial methods occasionally employed for purposes of advertising, as by fitting an under-size engine, overloading a chassis, maintaining the most efficient speed, etc.

Another engine of the same type installed in a similar chassis has run 70,000 miles and absolutely nothing was touched until 55,000 miles were reached, when a top overhaul was begun, but it was found that only valve grinding, carbon removal and cleaning of sprayers were needed. In both engines the lubricating oil is changed every 1800 miles and no "topping" has been found necessary between changes.

Two more engines of the same type have so far done over 30,000 miles each, and as yet have not been touched. A similar 4-cylinder engine rated at 68 hp. also installed in a G.M.C. chassis and used on the same run is hauling a pay load of 7 tons, with a fuel consumption of 14.2 miles per gallon. In these conversions the original rear axles have been retained, but an over-drive transmission has been installed to obtain the same road speed as with the original engines, in spite of a governed engine speed of 1700 r.p.m.

Fig. 5 shows a bus in which a 102 b.hp. engine of the same make has been installed. This unit has thus far run only 20,000 miles and it is therefore difficult to get correct operating data, except that the owners state that, with due allowance for future maintenance costs, they are expecting a net saving of 3 cents per mile, as compared with the original engine equipment.

A 6-cylinder Leyland engine of 4⅜ bore by 5½-in. stroke, developing 90 b.hp. (Fig. 6), installed in a tractor, has been in operation for 28 months and has covered over 110,000 miles. This unit, in combination with a four-wheel semi-trailer carrying about 24,000 lb., is used on a run of 480 miles total length, half of which is located in mountainous territory. At an average road speed between 23 and 25 m.p.h. the fuel consumption varies from 8.15 to 7.6 miles per gallon, depending mostly on weather conditions and average speed. The gross vehicle weight varies from a minimum of 44,000 to a maximum of 49,000 lb. The owner of this unit, who operates a large fleet and maintains an extremely accurate cost system, has found that the certain maintenance, overhaul and garage routine costs of the engine amount to a fraction under 4 cents per mile, whereas the same accounts on his gasoline vehicles of comparable size show a charge of 4.68 cents per road mile. The gasoline vehicles are rated at 125 hp. and, consequently, carry slightly heavier loads. They are, however, being used on an easier run. The lubricating oil consumption of the Diesel engine was found to be 8.4 gallons for 2595 miles. This operator uses a lubricating oil reclaimer and therefore changes his oil quite frequently, namely, every 1000 miles. During the total operation of the engine not a single road call, caused by the engine, was necessary and every run has been completed without delay.

A smaller truck of the same make, equipped with a 268 cu. in. engine (Fig. 7), carrying 15,000 lb. pay load, with a gross weight of 29,000 lb., has been operating for several months. The weekly mileage is between 1300 and 1500. It is being used on long runs, mostly through hilly country and not always on first-class roads. The average road speed is between 28 and 30 m.p.h. With a load of 15,000 lb. the fuel consumption is 12.95 miles per gallon, and with 4000 lb. 19.15 miles per gallon. The consumptions give 167.5 and 147.5 gross ton miles per gallon, respectively, which, considering the small size and the high average road speeds, is a remarkable performance. The owner, who keeps an accurate accounting system, has found that all costs, except fuel, interest and depreciation, which he figures on a time basis, are exactly the same as for his gasoline vehicles.

The average fuel cost in Ontario is 18.3 cents for gasoline and 13.3 cents for fuel oil, both these figures including the tax of 6 cents per Imperial gallon, or 5 cents per U. S. gallon.

All figures given refer to U. S. gallons and to tons of 2000 lb.

In February, 1933, the Montreal Tramways Company put into operation a 29-seater Leyland Diesel bus, weighing about 14,900 lb. This vehicle is equipped with a 6-cylinder engine of 496 cu. in. swept volume developing 90 b.hp. at 2000 r.p.m. governed speed.

D. E. Blair, General Superintendent of Montreal Tramways, reported on the performance of this unit to the Canadian Transit Association meeting in June, 1934. Since that time the vehicle has covered an additional 40,000 miles, and has thus operated over a total distance of approximately 115,000 miles.

Fig. 4—G.M.C. Truck Tractor Diesel Conversion with Gardner 6-LW Engine

The following abstract of Mr. Blair's report, which will be presented at the 1935 meeting of the Canadian Transit Association, contains complete data on the operation of this vehicle to date.

"Costs of the Diesel bus are compared with the average cost of five gasoline buses of identical pattern, except for the difference in the engine unit, for the 27 months of the test. The only items considered are those that would be affected by this difference. The gas buses are now between four and five years old.

	Diesel, cents	Gas, cents
Fuel, per mile	1.45	3.90
Lubricants	.15	.19
Maintenance, chassis	2.09	2.39
Depreciation	2.74	2.54
Interest	1.15	1.06
	7.58	10.08

Note: Cost of fuel oil has been taken as including a 6-cent tax over the whole period, although it was only applied in 1934.

There is, so far, no indication of any but minor maintenance troubles. These are as follows:

1. Pintles in Bosch injector nozzles required reseating after about 100,000 miles.

2. Timing chain which drives Bosch pump was renewed after about 100,000 miles.

3. Several springs on plungers of Bosch pump were broken. These have been improved in design and quality and should have longer life. Cost is negligible.

4. High-pressure steel pipes between pump and nozzles failed through fatigue. Shape has been improved to obviate concentration of stresses at couplings.

Injector nozzles have been cleaned at intervals of 3500 miles and valves required decarbonizing at about 12,000 miles.

Experience gained in the matter of the delicate adjustment of Bosch pump will probably raise these periods to 5000 and 15,000 miles respectively.

Bosch injection nozzles are now being set to open at about 90 atmospheres or 1300 lb. per sq. in.

The addition of a spring hub at center of clutch plate has eliminated clutch rattle.

Drive line and universal joints show no greater wear than in gas buses.

Savings should amortize cost of new engine in little more than one year."

These results prove that the opinion held by some observers of European conditions, namely, that the Diesel engine can only successfully compete with the gasoline engine as long as it is favored by rate of fuel taxation, does not hold good, as the Canadian tax rate per gallon is the same for gasoline as for Diesel fuel oil.

New operating experiences gained in Canada refer mostly to operation in colder temperatures than are generally encountered in Europe.

In winter the fuel oil must be entirely free from water, as this accumulates in the tank and causes trouble before it can be separated by the filter. When the vehicle is kept in a warm garage and then driven out into the cold, the water contained in the fuel tank is shaken into an emulsion with the fuel and when the temperature of the tank contents drops below the freezing point, a slush is formed. The fine ice particles traveling with the fuel oil seem to be deposited in the fuel pipe between tank and filter and finally clog this pipe over its full length. This condition may be so bad as to make it impossible to remove the ice by any other means than by thawing the vehicle out in a warm garage and draining the whole fuel system.

Again it was found that some otherwise very suitable fuel oils deposited their wax content in the fuel filter, transforming the filtering material into a perfectly oil-tight receptacle. If it is impossible to get a fuel which will not deposit wax when kept at a temperature at least 5 deg. below the lowest service temperature, the remedy is to install the filter in a position where it will be kept warm, but care must be taken not to overdo this, as otherwise vaporlock will develop during summer operation. Wax is not usually deposited in the fuel supply pipe between tank and filter.

Necessarily, the fuel must be as clean as possible, and care should be taken in handling it. The practice, occasionally recommended, of adding waste crankcase oil, gasoline or paraffin to the fuel, is to be condemned. If this is done, the supposed saving by "reclaiming fuel which would otherwise go to waste" will be more than lost by increased maintenance cost.

The physical and chemical specifications of fuel oils do not generally cover their ignition qualities. It is deplorable that no standard knock rating of fuel oils, similar to that given by the octane number for gasoline, has been generally introduced. Consequently, the operator has no other means of

Fig. 5—Oil Engine Conversion of Interurban Bus

ascertaining the suitability of fuel than by trial. A few of the oil companies selling a "light Diesel fuel" can give no information on the ignition and combustion qualities of their product and are often not even willing to state whether it is straight run, re-cracked or re-cycled. Using the proposed Cetene rating, the "Centene numbers" of "light Diesel fuels" on the market vary from about 35 to 75. This gives an indication of how much engine performance may change when different fuel oils are being used. Present high-speed oil engines are very sensitive to fuel characteristics, but designers are, of course, endeavoring to develop engines which can use a wider range of fuels, as has been done in the case of the gasoline engine.

With only some ten years of experience to draw from, the high-speed fuel oil engine is still in the early stages of its development. This is borne out by the fact that no stardardization of combustion chamber design is as yet noticeable. In comparing oil engine with gasoline engine performance the newness of the former should be taken into consideration, as one is inclined to measure the new design by standards set up by the older type, which has reached such a high degree of

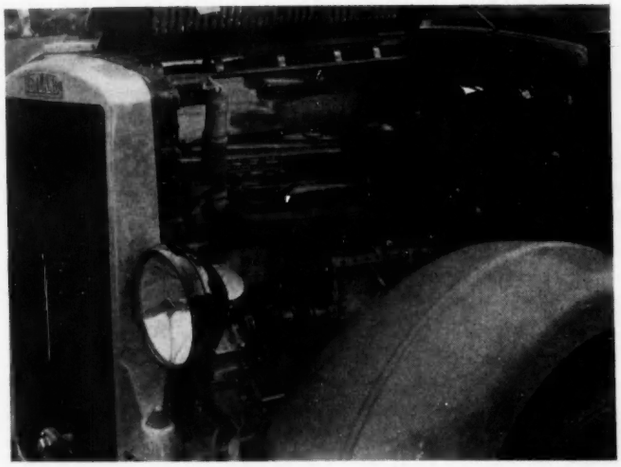


Fig. 6—496 Cu. In. Leyland Engine

development, that, during the last few years, improvements have been possible only in minor detail.

The desirable components of fuel oils are precisely the opposite to those of gasolines. In the former paraffins constitute suitable components, while antiknock aromatics of ring-type structure cause rough running. It thus appears that the molecular configuration of the hydrocarbons determines their combustion characteristics, with open-chain molecules lending themselves more easily to the formation of peroxides, which inaugurate ignition. In addition to short ignition lag, and to the absence of very slow burning components, the fuel oil has to possess a certain lubricating value to prevent undue wear in the injection pump. It is to be noted that a fuel which is perfectly suitable for one class of combustion chamber design is not necessarily equally suited to another. This will have to be considered when a standard method of fuel oil knock testing is decided upon.

On account of limitations of space it is impossible to consider in greater detail the question of fuels, and lubricating oil characteristics can only be discussed briefly.

It is well known that different types of oil engines require different grades of lubricating oil, depending on the varying

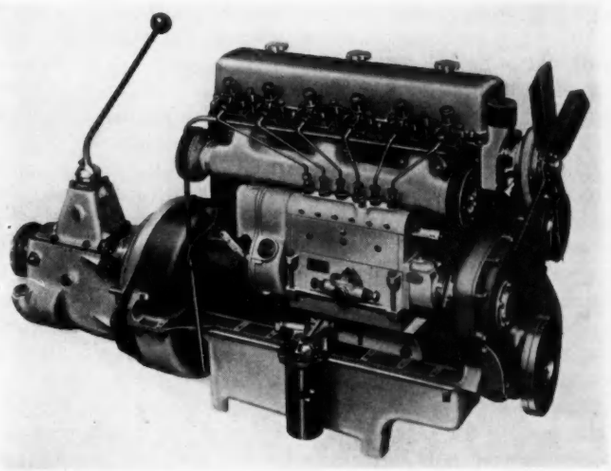


Fig. 7—Leyland "Cub" Oil Engine, 3⅜-In. Bore by 5-In. Stroke

amounts of dilution and on different temperature and pressure conditions. Generally speaking single-cell engines require a more viscous oil than multi-cell engines, although there are exceptions to this rule.

All Diesel engine oils must have high resistance to oxidization, carbonization, and emulsification. In cold weather operation a free flowing lubricating oil is even more important for compression ignition than for gasoline engines as, otherwise, starting may be rendered impossible. So long as the oil employed was such that the engine could be turned over reasonably freely, no starting difficulties were, to the author's knowledge, encountered by Canadian operators. The ease of starting in extremely low temperatures is particularly impressive in most direct injection engines, which, of course, have no auxiliary heating devices. Some of these engines which are equipped with special means to apply extra high compression for starting, use them rarely, if ever. The same applies to the compression release fitted to most direct injection engines except, of course, when no electric starter is employed. The drivers of engines which must be started by hand do not complain about this, and after a winter like that of 1934-35, this proves that starting really is easy.

Major overhauls of automotive Diesel engines were rather costly in the early days, as it was necessary to re-grind cylinders, fit under-sized big-end bearings, etc., more often than in gasoline engines. Better combustion control, special cylinder materials, nitrited liners, pistons with cast iron inserts to support the upper piston rings, hardened crankshafts, and new connecting rod and main bearing materials, or bearings of the anti-friction type, have eliminated this additional maintenance cost, although, as far as special materials are concerned, at the expense of the initial cost. Many of the early difficulties were due to maintenance men attempting to "improve" the injection equipment, and to insufficient attention being paid to smoky exhaust, which is always an indication of engine trouble. This latter characteristic of the oil engine has caused a considerable amount of adverse criticism, but, in the author's opinion, this is an advantage from the operating viewpoint, provided, of course, that the exhaust is free from smoke when the engine is in proper operating condition. After the operating staff has been taught the meaning of a smoky exhaust, and has learned not to attempt cleaning a faulty sprayer on the road, the cause of most of

the high maintenance cost has been removed. If no spare is carried it is preferable to drive home on the remaining, properly operating cylinders. It is quite unnecessary to make a Diesel expert out of every driver, but it is essential to impress on those drivers who are not skilled mechanics, that they must not meddle with things that they do not understand.

The maintenance staff has to be taught only a small number of things, among them, that misfiring is hardly noticeable when the engine is idling if a governor holds it to a steady speed. To test correct engine operation it is necessary to make the engine pull hard at about one-third of its maximum speed. It is impossible, within the scope of this paper, to enumerate all of the little tricks used in successfully operating and maintaining oil-engined vehicles, but it can truthfully be said that after a mechanic has grasped the fundamental principles of the Diesel engine he will have no more trouble in acquiring the necessary knowledge of the details than he had in learing how to time an ignition and how to adjust a carburetor.

It has been mentioned, already, that some of the experiences with early conversions were rather unhappy. Generally such disappointments are caused by insufficient knowledge on the part of the person entrusted with laying out the conversion work. As the business of converting existing chassis to oil engine drive has attained considerable proportions in Britain, as well as in other European countries, it may be expected that a similar business will spring up on this continent. This business will probably be carried on by local agents of the engine manufacturer. It will be advisable not only to pick these agents very carefully, but also to supply them with all the parts required to make a complete conversion. They should not be expected or allowed to evolve by themselves brackets, connections, etc., as any difficulties, occurring after the converted vehicle is put into operation, will be blamed on the engine maker. How serious this problem can be is best illustrated by a few instances.

The author has seen a front chassis cross member cut away, to accommodate the Diesel engine crankcase, to such an extent

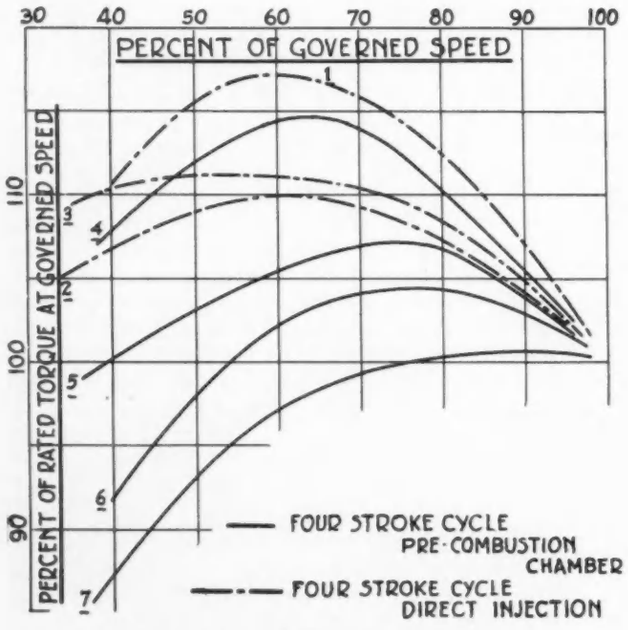


Fig. 8—Torque Curves of Some Direct Injection and Pre-Combustion Chamber Diesel Engines

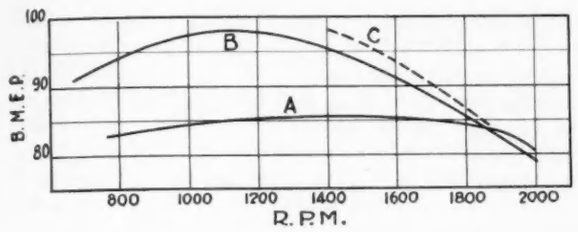
that it failed to stiffen the chassis, and the engine crankcase, which was not designed for this purpose, had to take over the function of the cross member, with obvious results. Rear engine supports were riveted or bolted to the frame by the trial-and-error method, leaving the frame members full of drilled holes. An engine was mounted "flexibly" in such a fashion that vibrations were set up which made it impossible to keep one's hands on the steering wheel. The original gear box was mounted on the new engine by means of an adapter ring which lasted only a short time. The stump shaft of the clutch had no provision for lubrication. Engine accessories, such as exhauster or compressor, when found to interfere with some fixed part, as for instance the steering gear, were mounted in some new position which made it impossible for them to function properly.

It is also advisable to prescribe the engine type to be used to replace a certain gasoline engine, and not to leave it to the discretion of the agent to make his own choice merely from the viewpoint of equal rated brake horsepower. This involves the consideration of a fallacy which has been propagated ever since the first automotive oil engine catalog was printed. It has been, and still is, contended that the oil engine, by virtue of its "well sustained" torque characteristic (meaning a nearly constant torque over the full speed range), is better suited to vehicle drive than is a gasoline engine. The truth is that exactly the contrary would be the case if the torque were really constant, and the oil engine would give rather a poor performance if it were not for the fact that it responds more promptly than does the gasoline engine when the accelerator is depressed. The latter statement holds good only if this quick response is not nullified by a sluggish governor.

In selecting an engine to obtain a given performance it is necessary to pay close attention to the torque characteristic. There is very little variation in the torque of gasoline engines of different design, generating the same horsepower at the same speed, and doing the same kind of work, but this is by no means the case with compression-ignition engines, as is shown by Fig. 8. The torque curves of a number of engines of different makes and approximately the same rated brake horsepower at similar governed speeds have been plotted in percentage of rated torque at governed speed. These curves show that the *maximum torque* may vary as much as 20 per cent and the vehicle performance will be affected accordingly. Although suitable torque characteristic for vehicle drive is sometimes considered to be obtained by direct injection engines only, it should be noted that Curve 4, pertaining to a pre-combustion chamber type engine, is very close to Curve 7, which pertains to a direct injection engine. It is also interesting to note that Curves 4 and 5 represent engines made by the same firm, both being of identical basic design, but No. 4 belongs to an engine put on the market in 1934, whereas No. 5 relates to one brought out in 1930. The change in torque characteristic proves that this manufacturer understands the requirements of automotive drive.

The performance of an oil engine will, however, improve in service relatively to that of a gasoline engine. Whereas in the latter case it is left to the discretion of driver or garage foreman to decide when an overhaul or check-up is necessary and therefore the engines are often still being operated when they should already have undergone a repair, the oil engine will, by smoking, make an inspection imperative. Thus, it either operates efficiently or not at all.

When selecting an engine it is necessary to ascertain



A - ENGINE PERFORMANCE WITH FIXED LIMIT TO FUEL PUMP OUTPUT
B - ENGINE PERFORMANCE WITH VARIABLE FUEL PUMP STOP
C - SMOKE LIMIT

Fig. 9—Improvement of Engine Performance by Varying Maximum Fuel Pump Volume in Accordance with Engine Speed

whether the published power curve was obtained with or without a fixed stop on the fuel pump which limits the maximum pump output. It is evident that, as a stop is always used in service, only the power developed with this part in place is of any interest, but some engine makers do not seem to realize this. To the author's knowledge, all power curves, issued now, represent the engine output with clear exhaust, unless otherwise specifically stated, but the question of pump stop seems to be overlooked occasionally. Disregarding smaller influences such as changes of turbulence, resistance of antechamber or turbulence chamber throat, mechanical efficiency, etc., the torque changes, at various engine speeds, depend primarily on two factors: volumetric efficiency of cylinder and volumetric efficiency of pump. In the average engine the former decreases above approximately 1000 r.p.m. In most fuel pumps, at least of the type where the pump piston controls the intake, volumetric efficiency increases with increasing speed. A fixed stop has to be adjusted in such a fashion that the pump output is limited to that quantity of fuel which can be burned at maximum engine speed, when the amount of available oxygen is a minimum. The fixed pump stop will, therefore, hold the engine at all speeds lower than the maximum to a lower torque than can be produced without a stop. One maker has remedied this difficulty by means of pressure controlled bypass valves in conjunction with a restricting orifice inserted in the injection pipes. The same result can, however, be obtained by very simple mechanical means and without interfering with the hydraulic part of the injection equipment. Fig. 9 shows, in Curve *A*, the torque of a turbulence chamber engine as furnished by the maker, and in Curve *B*, the gain in performance obtained by a mechanical device, altering the maximum pump output in accordance with engine speed. Curve *C* indicates the smoke limit. The maximum cylinder pressures were not increased by this alteration.

Except for the above, which applies only to a few manufacturers, there is very little fault to be found with the data given out by oil-engine makers. So far as their products are concerned, some question might, however, be raised.

It is one of the inherent advantages of engines with internal mixture formation that equal output of all cylinders *should* depend solely on proper adjustment of the fuel pump and nozzles. Why, then, some makers adopt an intake manifold similar to that used on gasoline engines is hard to explain. If, in spite of a manifold, the weight of aspirated air is the same for all cylinders, no objection can be raised and the advantage is gained that only one air cleaner is necessary. Sometimes, however, individual cylinders smoke at certain speeds in spite of perfectly equalized injection equipment, and if all cylinder parts are found to be in perfect working order, the bad performance must be blamed on the effect of the manifold.

Insufficient attention is being paid by some proprietary engine makers to the location of normal chassis parts, as, for instance, steering gear or steering column, location of pedals, etc. This is natural if the engine was built in a country where the vehicle control is located on the opposite side, but should not happen where left-hand control is used in the country of origin. In producing new models engine builders should consider the reappearance of forward-control vehicles. What can be done in this respect is shown by Fig. 10 for an engine designed for right-hand vehicle control.

Where a compression release or similar device is used, it would be helpful if it were built onto the engine in such a fashion that it could easily be connected for operation from the driver's seat. The use of bellcranks, rods, etc., could then be avoided. If a high compression ratio for starting is provided, it should be made impossible to operate the vehicle with the engine in extra high compression. To make the compression lever inaccessible from the driver's seat does not do this, as even when it is located under the hood the driver might forget to reset it to running compression after the engine has started.

It is customary to crank hand-starting engines with their compression released and to re-establish compression only after the flywheel has accumulated sufficient energy to overcome two or three compression strokes. There does not seem to be any reason why a similar procedure should not be followed where an electric starter is being used. The compression release could easily be made to operate in conjunction with the starter, and starter size and battery discharge could be reduced considerably. This is being done in some electrically starting marine engines.

Reference has already been made to sluggish governors and their effect upon acceleration. It remains to be said that the governor must also be quick in action to make possible fast gear shifting without clashing and without losing most of the momentum of the vehicle. What could be gained from more responsive governors can best be observed by driving any oil-engined vehicle, equipped with variable speed governor control, with accelerator directly connected to the pump control. In most cases the difference in performance is material.

In quite a few instances, particularly of Continental European engines, cooling water temperature control could be vastly improved. Insufficient attention has, as yet, been paid to bringing the engines quickly and evenly to a proper operating temperature. This is more important for an oil engine

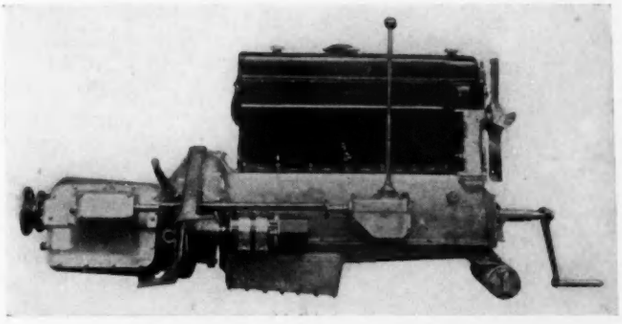


Fig. 10—Vehicle Control Side of Leyland 524 Cu. In. Engine

than for the gasoline type, as the former will pull when too cold, although with considerable knocking, which is caused by too low a compression temperature resulting in increased ignition delay. A driver who is in a hurry to get away is liable to disregard this knocking, and thus cause increased maintenance cost. In this way the smaller amount of heat transferred to the cooling water, which otherwise is advantageous, may prove to be a drawback, if proper arrangements for temperature control are not made. A throttle-type thermostat in the cooling water discharge does not constitute a remedy, particularly if the radiator is too large, as is always the case when, in a conversion job, the original gasoline engine radiator is retained. A thermostat of the bypass type *must* be used, but this does not seem to be appreciated by some makers.

An argument often used against the oil engine is its low mean effective pressure in comparison to the gasoline engine. The difference, particularly after a few hundred miles of operation, is not as great as is generally contended. A unit recently brought out by a European maker—who had previously built engines of the original air cell and cross-stream types—is developing a peak of 112 lb. b.m.e.p. at 1000 r.p.m., and better than 100 lb. b.m.e.p. over the range from 400 to 1400 r.p.m., using a direct injection system. These figures, referring to performance with clear exhaust and normal fuel pump setting, prove that the oil engine no longer lags far behind its gasoline ancestor.

To obtain a fair comparison of weights, this should not be based on engine weight only, but on the total weight of engine, radiator with contents, tank and fuel necessary for a given run. This last item will, for the average 100 hp. truck and for a 450-mile run, amount to something like 250 lb. to the credit of the Diesel if, according to our operator's custom, sufficient fuel is carried for the whole trip. This is, of course, an unimportant percentage of the gross vehicle weight but should be considered when the question of respective engine weights is being argued.

The endeavor of engine designers will naturally be to increase the mean effective pressure but this cannot be increased indefinitely. As ignition pressures and volumetric cylinder efficiencies are already as high as can be hoped for—without employing supercharging—the greatest benefit is to be expected from further increases in combustion efficiency, or, in other words, from reduction of excess air. It cannot be expected than an engine employing internal mixture formation will ever be able to use the oxygen content of the cylinder charge as completely as will a gasoline engine, but most of the former are still short of the oxygen utilization of 90 per cent shown by one or two designs. It can reasonably be predicted that single-cell engines, or some engines of the auxiliary air chamber type, can more easily be brought to this state of perfection than engines of the pre-combustion chamber or turbulence chamber type.

One British maker has had supercharged bus engines on the road for a considerable time and their performance is reported to be most satisfactory. Other important steps forward are air-cooled automotive oil engines, which are of particular importance for military use, and some excellent results obtained with two stroke cycle engines.

During the last two years the oil engine has made rapid advances in Canadian road transport. That this is not looked upon as a passing fancy is proven by some oil companies which have purchased Diesel equipment for their fleets. This action can hardly be construed as an effort to advocate the use of this type of engine to the detriment of gasoline sales, but must have resulted from the conviction that this type of equipment has a future. Instead of trying to retard this development, these oil firms take reasonable action by acquiring first-hand knowledge of the suitability of their fuels.

It is impossible even to estimate the total number of oil engines in operation on Canadian roads, but it is sufficient to state that one firm alone has sold over 4000 hp. of automotive oil engines during the last six months. This figure may not be very impressive, measured on a scale of American truck engine production, but represents a relatively large part of the heavy vehicle engine sales in Canada during this period. Unless respective fuel costs should be changed materially from their present levels, a further increase in the ratio of oil engine to gasoline engine sales can reasonably be expected. This prospect will be improved if freight shipments should continue to increase and if some of the smaller operators should thus regain a position which would enable them to invest a little more money in first cost of equipment, or if some of the oil engine sales organizations should adopt time payment plans extended over as long a period as those used by some gasoline truck manufacturers.

The author wishes to express his thanks to D. E. Blair and to the other operators who kindly permitted him to cite their operating experiences, as well as to Prof. E. A. Allcut, University of Toronto, for his many valuable suggestions.

MEETINGS CALENDAR

Production Meeting
Sept. 18 & 19
Hotel Statler, Cleveland
(In conjunction with Machine Tool Congress Sept. 11-21, 1935)

Transportation Meeting
Oct. 10
Palmer House, Chicago

Tractor and Industrial Power Meeting
Oct. 11 & 12
Palmer House, Chicago

S.A.E. Annual Dinner
Nov. 4
Hotel Commodore, New York City

S.A.E. Annual Meeting
Jan. 13-17, 1936
Book-Cadillac Hotel, Detroit

Section Meetings

Metropolitan—Sept. 20

Westchester Country Club, Rye, N. Y., 2 P. M. Bridge, golf, dinner and dance.

Northern California—Sept. 10

Engineers Club, San Francisco; dinner 6:30 P. M. Relationship of Piston Rings to Performance—D. D. Robertson, chief engineer, Wilkening Manufacturing Co.

Southern California—Sept. 7

Western Dairy Products Farm, Elmonte, Calif. Sports in the afternoon, barbecue steak dinner at 6 P. M., followed by technical session, at which D. D. Robertson, chief engineer, Wilkening Manufacturing Co., will talk on Piston Rings and Their Relation to Motor Operations.

Non-Ferrous Metal Specifications Get Thorough Revision

By Paul V. Faragher
Metallurgist, Aluminum Co. of America
Chairman, Non-Ferrous Metals Division, S.A.E. Standards Committee

THE widely used S.A.E. non-ferrous metals specifications have just been carefully revised and brought up-to-date in every particular. Most significant are the changes which have been made in the specifications for aluminums, brasses and bronzes, but each item in every category has been reviewed in detail and revisions have resulted all along the line.

Every year design and production developments in the automotive industry have combined to increase the vitality of this set of specifications. Their practical value has grown consistently ever since non-ferrous metallurgy took its first important automotive step in 1905 when standard specifications for babbitts and valve metals were adopted by the Mechanical Branch of the Association of Licensed Automobile Manufacturers, forerunner of the S.A.E.

During the S.A.E.'s first year, additional specifications were added to the standard and included solders, new babbitts and a variety of copper base alloys both for ingots and fabricated materials such as rods, sheet and wire. Later, as the use of aluminum in the automotive industry became more general, the first few aluminum specifications were agreed upon and adopted. As this metal became more widely used, demands arose for alloys with particular characteristics such as high strength for aircraft and other important applications, and as rapidly as they were developed, standard specifications were agreed upon and adopted.

As die castings came into use in automobile fabrication, zinc alloy specifications were adopted and several valuable informational notes were published that include nickel and chromium plating, tungsten, platinum, tungsten carbide, etc.

The specifications include chemical analyses, normal physical characteristics, notes on general uses and special information and in the case of fabricated materials, dimensional specifications and tolerances. The general groups, especially the aluminum alloys, the brasses and bronzes are supplemented by a wealth of general information that is very valuable to users having limited metallurgical laboratory facilities.

All the specifications were formulated by Committees composed of producers and users of these metals and were reviewed by the industry at large before their adoption. Their

Paul V. Faragher

reliability and general acceptance have been evidenced by their general use in many mechanical industries other than the automotive.

All of the revised specifications are included in the current S.A.E. HANDBOOK which is available at $5 per copy. A wide demand for the non-ferrous metal specifications has led the Society to publish them also separately in a complete strongly covered booklet that is obtainable at $1.50 per copy. All inquiries should be directed to the Society at its offices in New York City.

The following listing shows briefly the scope and extent of these specifications:

Classification	*Compositions*	*Present Specification Numbers*
Solders		
	Classes A and B	1-4
White Bearing Metals		
	Cast and Ingot	10, 12-14
	Cast and Ingot	11
Wrought Aluminum		
	Alloys, Notes on	
	Type 1 Alloys	20, 29
	Type 1 Alloys	25
	Type 2 Alloys	24, 26-28
Cast Aluminum		
	Alloys, Notes on	
	Type I Alloys	30-33, 31A, 320, 35, 36
	Intermediate Alloy	37
	Type 2 Alloys	321-324, 34, 38, 39
	Die Casting Alloys	304, 305, 307, 309, 312
Cast Brass and Bronze Alloys		
	Notes on	
	Red Brass	40
	Yellow Brass	41
	White Nickel Brass	42
	Manganese Bronze	43
	Cast Brass to be Brazed	44
	Brazing Solder	45
	Hard Cast Bronze	62
	Leaded Gun Metal	63
	Phosphor Bronze	64
	Nickel Phosphor Bronze	640
	Phosphor Gear Bronze	65
	Bronze Backing for Bearings	66
	Semi-Plastic Bronze	67
	Aluminum Bronze	68
Wrought Copper and Copper Alloys		
	Alloys, Notes on	
	Commercial Brass Sheet	70
	Aluminum Bronze	701
	Copper Sheet	71
	Annealed Seamless Brass Tubing	74
	Copper Tubing	75
	Naval Brass (Tobin Bronze) Tubing	76
	Phosphor Bronze Strips	77
	Red Brass Sheet	79
Wires and Rods		
	Free Cutting Brass Rod	72
	Naval Brass (Tobin Bronze) Rod	73
	Brass Wire	80, 82
	Phosphor Bronze Wire	81
	Soft or Annealed Copper Wire	83
	Brass Rod	88
Zinc Alloys		
	Die Castings	903, 921
Valve Metals		
	Nickel Iron	
Miscellaneous Alloys		
	Notes on	

Standardization of Speedometers, Odometers and Tachometer Drives

IN response to requests from several sources, the Society has undertaken a program leading to the standardization of speedometer and odometer calibrations (a new subject) and drive-shaft standardization (possible revision of the S.A.E. Standard for tachometer drive-shaft connections, page 26, 1935 edition S.A.E. HANDBOOK).

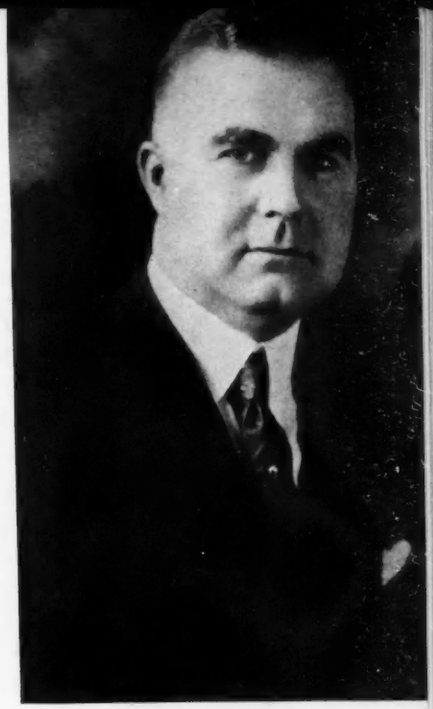
The project has been referred by the S.A.E. Standards Committee to a Speedometer Subdivision of the Parts and Fittings Division of which F. G. Whittington, Moto Meter Gauge & Equipment Corp., is chairman.

At an informal meeting to consider the scope of the project, agreement was reached upon eight principles to form the basis for a standardization.

With respect to drive-shaft standardization it was pointed out that it does not seem feasible at the present time to establish a single universal standard. One suggestion made was the possibility of continuing the present S.A.E. aeronautic type of drive-shaft connections, for light duty and to substitute square shaft tips for heavy-duty, marine and similar applications.

Other members of the Subdivision in addition to Mr. Whittington are: W. H. Hutchins, Delco-Appliance Corp.; P. J. Kent, Chrysler Corp.; A. W. LeFevre, Stewart-Warner Corp.; C. W. McKinley, AC Spark Plug Co.; F. G. Gardner, Breeze Corps., Inc.; and E. T. Blanchard, Barbour-Stockwell Co.

Leading Engineers Praise Revisions Preserving Vitality of S. A. E. Standards



Jesse G. Vincent
Vice President of Engineering, Packard Motor Car Co.

says

". . . Reliable Guide in Selection of Materials"

THESE non-ferrous metal specifications, together with the accompanying information, have received wide recognition in the automotive industry and afford a most reliable guide for the selection of suitable materials for automobile parts.

They now cover the range of present-day materials but still offer the necessary simplification.

Robert G. N. Evans
Engineer in Charge of Research, Bunting Brass & Bronze Co.

says

". . . Recognized Throughout the Industry"

THE greatest value in the S.A.E. specifications lies in the fact that they are recognized as standards throughout the automotive industry and are now being specified by nearly every plant manufacturing machinery.

Today when a metal is recommended by our engineering department for any specific application, we are asked, "Does it carry an S.A.E. number?" and if so, the recommendation is accepted without further discussion.

Fred M. Zeder
Vice-Chairman of the Board, Chrysler Corp.

says

". . . Needs of the Buyer are Filled"

THE value of the S.A.E. non-ferrous metal standards is that they represent the combined efforts of both producer and consumer. They reduce the findings of years of experimental and development work to ready reference form.

One of the dangers or disadvantages of specifications is that once written they are regarded with too much respect. They are held as too unchanging—and not altered to meet constantly changing requirements. This danger is avoided by periodic revision by the standards group responsible as is the case in connection with S.A.E. standards in general and these non-ferrous specifications in particular.

Credit should be given to those technologists who are responsible for such standards.

Charles B. Bohn
President, Bohn Aluminum & Brass Corp.

says

". . . Useful for Frequent Reference"

THE S.A.E. non-ferrous metal specifications are very useful to the entire industry, both because of their practical character and because they are presented in a clear and accessible manner.

Our metallurgists and sales engineers make frequent reference to them in various phases of their everyday work.

What Members Are Doing

Paul W. Litchfield, president, Goodyear Tire & Rubber Co., arrived in New York, July 25, after a trip around the world which included visits to six of his company's plants. Mr. Litchfield's return coincided with his sixtieth birthday and with the anniversary of his thirty-fifth year with the company, which he joined in 1900 as superintendent.

Paul W. Litchfield

John H. Geisse, chief of the development section of the Bureau of Air Commerce, piloted a Waterman tailless airplane from Santa Monica, Calif., to Washington, arriving at the latter city Aug. 12. The new plane is one of four experimental types purchased by the Bureau under a program of investigating possibilities in planes for private ownership. The section which Mr. Geisse heads has been responsible for the project and the transcontinental flight with the new plane gave Mr. Geisse an opportunity to prove to his personal satisfaction that it was "nearly foolproof".

Bernt Balchen, formerly pilot with the Ellsworth Transantarctic Expedition and the Byrd Expedition, is now with the Det Norske Luftfartselskap Fred. Olsen & Burgenske A/S., in Oslo, Norway.

Dr. Jerome C. Hunsaker, head of the mechanical engineering departments at the Massachusetts Institute of Technology, has been named technical adviser to the United Aircraft Manufacturing Corp., recently organized parent corporation for the manufacturing divisions of the United Aircraft Corp.

Miss Amelia Earhart has resigned her connection as dollar-a-year expert of the Bureau of Air Commerce to devote herself to her commercial-aviation activities. During her connection with the bureau she made practical tests of a direction-finding compass.

Walter A. Olen is celebrating his twenty-fifth year as president and general manager of the Four Wheel Drive Auto Co.

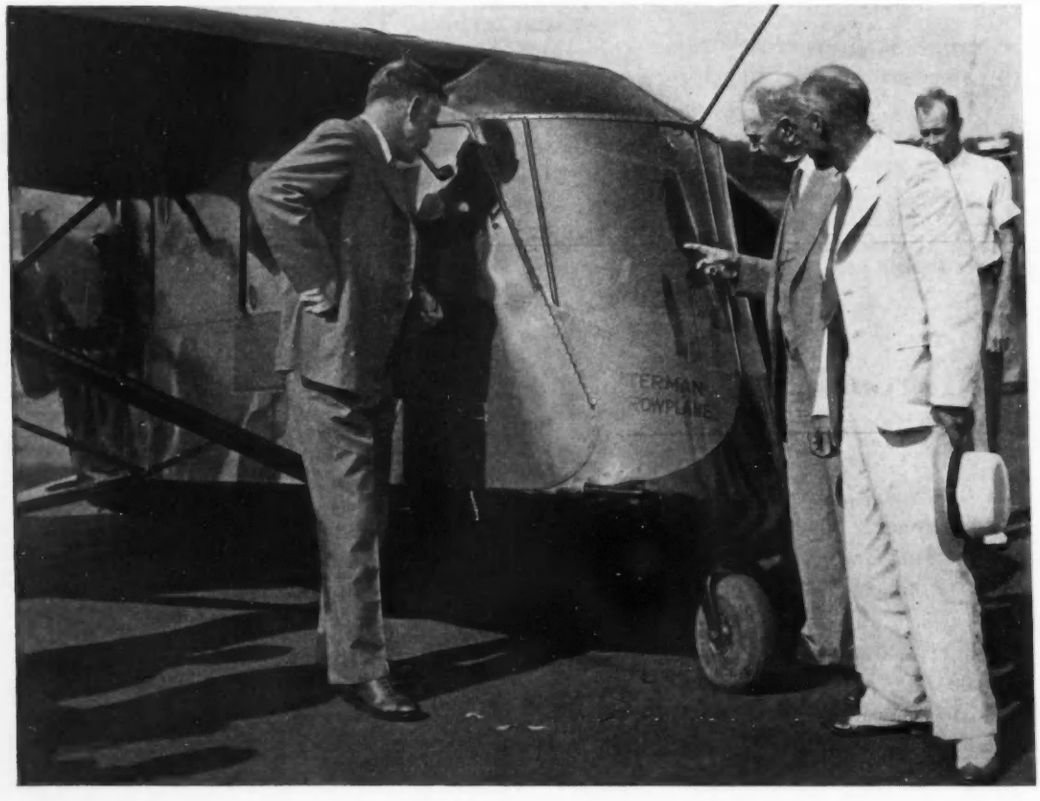
Floyd F. Kishline has been named chief engineer of the Graham-Paige Motors Corp., having previously served as assistant chief engineer of the company. In 1934 he was vice-president of the S.A.E. representing the Passenger Car Engineering Activity and has been identified with many of the Society's Committees. In his new position he succeeds *Louis Thoms,* who has resigned.

Charles W. Jinnette, for the past 25 years district manager in Detroit for the abrasive division of the Norton Co., has been appointed regional manager for sales planning and development of the company's operations in the Middle West. He will continue to make his headquarters in Detroit.

R. H. Combs, president, Prest-O-Lite Storage Battery Co., Ltd., has been elected a member of the council and executive committee of the Association of Professional Engineers of Ontario.

Arthur W. Soutter is now employed as research engineer by the Gilbert & Barker Manufacturing Co. of West Springfield, Mass. He was formerly with the Springfield Manufacturing Corp., East Springfield, as chief engineer.

Geisse Flies Tailless Plane Across Country



Just after landing the tailless airplane at Washington, John H. Geisse (right), was met by Secretary of Commerce Roper and J. M. Johnson, Assistant Secretary of Commerce, who showed great interest in the construction of the airplane

E. F. Roberts, vice-president in charge of manufacturing of the Packard Motor Car Co., is retiring from active business after nearly 40 years in the automotive industry. He plans to travel for a year or two. His association with Packard dates from 1903, when he went to Warren, Ohio, to work as a tool maker in the plant where the first Packards were built.

E. L. Mench, Jr., formerly with General Motors Truck Co. in a sales capacity, is now assistant transportation engineer, International Harvester Co. of America.

Horace A. Brown, Jr., is now district sales manager, The Cleveland Welding Co., located in Detroit. He was formerly manufacturer's agent in Detroit.

D. F. Geisey

D. F. Geisey has joined the Harrisburg branch of the Four Wheel Drive Auto Co. as technical sales assistant.

Alexander W. Freyman is now with the Godfrey Manufacturing Co., New Brunswick. He was formerly layout draftsman, Wright Aeronautical Corp., Paterson, N. J.

Frank G. Born has severed his connections with Delco Products Corp., Dayton, Ohio, and is now chief engineer, Warner Electric Brake Manufacturing Co., Beloit, Wis.

Otto Mueller is now with the Toledo Machine & Tool Co., Toledo, Ohio. He was formerly connected with the Murray Corp. of America, Frame Division, Ecorse, Mich.

Temple N. Joyce
Recently named president of the Bellanca Aircraft Corp., New Castle, Del.

George E. Denis is now associated with Fairbanks, Morse & Co., Beloit, Wis., in their Diesel engineering department. He was formerly chief engineer for Universal Motor Co. of Oshkosh, Wis., for several years and more recently in charge of the engineering department, Young Radiator Co., Racine, Wis.

Christopher S. Oliver, works manager, Armstrong Siddeley Motors, Ltd., Coventry, England, was a visitor recently at the Society's headquarters in New York. On leaving New York he planned to visit automobile and aircraft plants in Detroit, Canada, and on the eastern coast of the United States. Paramount among the interests in his field, Mr. Oliver noted, were light alloys for aircraft engineparts.

Joseph Geschelin, having been transferred from Philadelphia to Detroit as technical editor of *Automotive Industries,* has resigned as vice-chairman of the Philadelphia Section of the Society.

J. T. McGuire, formerly assistant master mechanic, Chrysler Corp., Highland Park, Mich., is now superintendent, Highland Park Plant.

Louis P. Prossen, formerly president, Prossen-Meder Corp., New York, is now connected with the Studerus Oil Co., Inc., Kearny, N. J., as engineer on industrial fuel burning equipment, furnace designs and combustion engineering.

Capt. Vernon Peterson has been named district inspector, headquarters, Fort Douglas District, Civilian Conservation Corps. He was formerly commanding officer, Civilian Conservation Corps, Co. 1965, Manila Camp DF-35, located at Green River, Wyo.

Dr. Edward Sokal, president, American Katalite Corp., is temporarily located at 297 E. Grand Boulevard, Detroit, Mich.

H. C. Olivier, consulting engineer, a foreign member of the Society resident at Eindhoven, Holland, has published recently a thesis which is a critical study of present-day floating-power constructions, with a study of means of eliminating torque reaction from the automobile frame. The thesis, the title of which is, in Dutch, *Bescherming van het Onderstel Eener Zuigermachine Tegen de Veranderlijke Reacties van het Draaimoment,* was presented to the faculty of the Delft Technical School in fulfillment of the requirements for a doctorate.

Paul S. Bigby, formerly research engineer, Phillips Petroleum Co., Bartlesville, Okla., is now in charge of city sales in St. Louis for the company.

Section Statistics Reported at Semi-Annual Meeting

ONE HUNDRED AND SIXTY-EIGHT Section meetings were held from September, 1934, to June, 1935, at which there was a total attendance of 22,677. This compares with 150 meetings held during the same period last year, with an attendance of 19,000, representing an increase of 18 meetings and 3,677 in attendance over last year. This was brought out in a report presented to the Sections Committee at a meeting held during the Summer Meeting of the Society at White Sulphur Springs last June. Other facts brought out by the report were:

An increase in Section meeting attendance has been very noticeable in nearly all Section territories, and the Section officers are to be congratulated on the fine programs scheduled to warrant this increase in interest on the part of the Section membership.

Fifteen regional meetings were held with very satisfactory results, a definite outcome of such activity in Hartford, Conn., being an application for a new Section of the Society, which is being placed before the Council.

30 ANNIVERSARY

S. A. E.

Pacific Regional Meeting

San Francisco, Calif.—Nov. 18 and 19
Hotel Whitcomb

Auspices of the S.A.E. Pacific Coast Sections

Papers and plant visits of interest to members in the fuels & lubricants, tractor, Diesel and transportation & maintenance fields

S.A.E. Professional activities cooperating on program

If you are going to be on the Pacific Coast at the time of the meeting—if you are a Pacific Coast member—

Plan to attend!

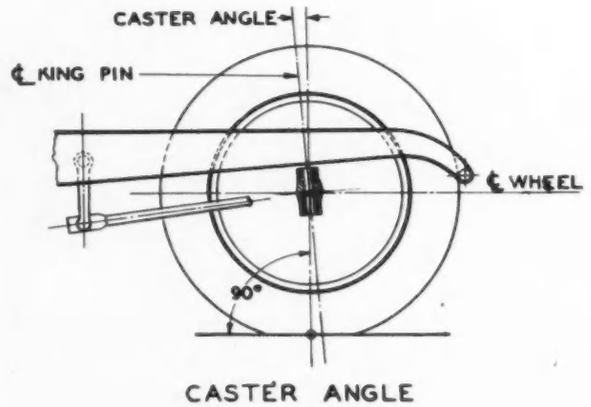
Wheel Alignment Elements Defined by Research Subcommittee

As a result of work begun by the Front Wheel Alignment Research Subcommittee in 1933, the Research Committee at its last meeting authorized the publication of basic definitions of wheel alignment elements covering caster, camber, toe-in and king pin inclination. B. J. Lemon is chairman of the Front Wheel Alignment Subcommittee.

These definitions have been developed for the purpose of making available to all interested in wheel alignment inspection and service, in broad terms covering all types of vehicles, definitions that may become standard and serve, until modified, as educational and workable guides.

Wheel alignment may be considered as the primary element of roadability of a motor vehicle and as such is of paramount importance to the operating safety of the vehicle throughout its period of service. It is, therefore, a matter of responsibility of equal importance to the manufacturer, maintenance organization and the vehicle owner. The following definitions apply to new and used cars alike, when tire pressures are equalized.

Caster

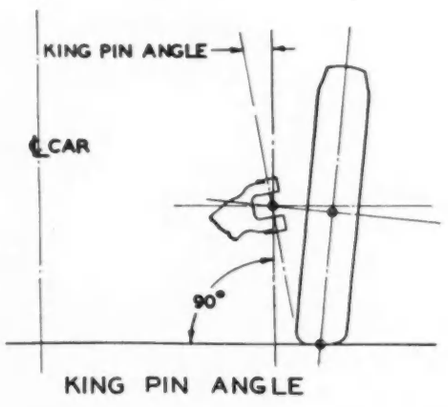


The caster angle of the steered wheels shall be the fore and aft inclination of the axis of the king or knuckle pin with respect to a transverse plane normal to the ground line of the vehicle. Positive caster shall mean that the projected axis of the king or knuckle pin meets the ground plane to the front of a wheel radius normal to the ground line.

On cars using the Dubonnet type of independent suspension, the king pins are not inclined fore and aft, but in a vertical plane at right angles to the centerline of the car. The wheel to ground contact point, however, is to the rear of this plane and its distance from the plane varies with load conditions more than on a leaf spring suspended car. The caster is given in fractions of inches rather than angle.

Caster angle of passenger cars, unless otherwise specified, shall be checked with the vehicle at curb weight[1] and manufacturer's specifications shall be based on vehicle curb weight. Manufacturers of commercial vehicles should specify caster angles at curb weight and at rated gross vehicle weights for standard and optional tires. Manufacturers should furnish tolerances for the above conditions of load and tire size and the caster angle should be within the manufacturer's limits.

King Pin Inclination

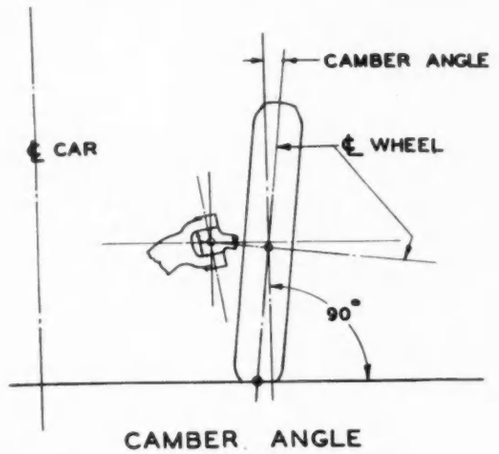


King or knuckle pin inclination shall be the in and out inclination of the axis of the king or knuckle pin with respect to longitudinal vertical plane. King pin inclination shall be checked with vehicle at curb weight, on a level surface.

Manufacturers should furnish tolerances affecting king pin inclination, and king pin inclination should be within manufacturer's specified limits.

[1] Curb weight shall be construed as the weight (resting on wheels) of the vehicle with its full complements of gasoline, oil, water, tools and regularly released accessories, but no passengers.

Camber

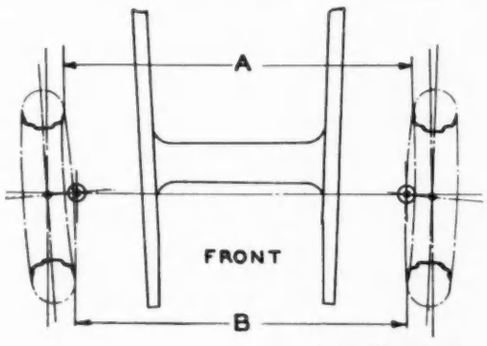


CAMBER ANGLE

The camber of the steered wheels shall be the inclination of the plane of the wheel with respect to a longitudinal plane normal to the ground plane of the steered wheels. Camber of steered wheels or vehicles using independent suspension or flexible axles shall be measured and specified with the vehicle at curb weight and with the transverse axis of the sprung mass[2] parallel with the ground line.

Manufacturer's specifications should be such as to provide tolerances sufficient to allow for maximum allowable lateral wheel and tire runout, king pin clearances, knuckle pin, knuckle pin bushing and wheel bearing wear. The camber of the vehicles should be within the manufacturer's limits.

Toe-In



TOE-IN

Toe-in shall be construed to mean that the planes of rotation of the steered wheels of the vehicle shall converge at a line in front of the vehicle and shall be the difference of the distances between the front and rear of the tires at a horizontal diameter. Toe-in of vehicles using an independent suspension or flexible connection shall be measured at curb weight and with the transverse axis of the sprung mass[3] parallel to the ground line. The toe-in should be within the manufacturer's specifications.

Wheel and tire runout shall mean the total sidewise variation of the tire relative to a fixed point on the ground line, checked at the maximum width of section of the tire.

Inspection Apparatus Approval

Inspection apparatus shall be of such design as to fulfill the requirements of wheel alignment. Approval of satisfactory inspection apparatus may be obtained from the Underwriters' Laboratories.

The foregoing basic definitions have been so formulated that their use does not require a particular type of instrument but defines the requirements of satisfactory instruments.

As has been the practice for several years past, the Front Wheel Alignment Research Subcommittee, with the approval of the Research Committee, has collected and here publishes wheel alignment specifications for 1935 passenger cars. In the past, these specifications have been of considerable benefit in improving alignment conditions by making this information available to independently operated service stations and others interested in checking or correcting alignment of passenger cars.

1935 Wheel-Alignment Data Passenger Car

Car	Caster, Deg.	Camber, Deg.	Toe-In, In.	King Pin Inclination, Deg.
Auburn				
653	3½ - 4	1½	⅛ - 3/16	7½
851	2 - 3	1½	⅛ - 3/16	7½
Buick				
40	2¾ - 3¼	0 - ¾	0 - ⅛	3½ - 4½
50	1¾ - 2¼	0 - ¾	0 - ⅛	4½ - 5½
60	1 - 1¼	0 - ¾	0 - ⅛	4½ - 5½
90	1 - 1¼	0 - ¾	0 - ⅛	4½ - 5½
Cadillac				
V-8	1 - 2	0 - ½	0 - ⅛	4½
V-12, V-16	1 - 2	0 - ½	0 - ⅛	4½
La Salle	1 - 2	¼ - 1	0 - ⅛	4 5/6
Chevrolet				
Master (Ind.)	0[a]	¼	1/16 - 3/32	7¾
Standard	1¾ ± ½	1 ± ½	5/64 - ⅛	7⅙ ± 1
Master (Conv.)	2¾ ± ½	1 ± ½	5/64 - ⅛	7⅙ ± 1
Chrysler				
Airflow CW	1 - 3	¼ - ¾	0[b] - 1/16 (Wee Gee)	10 ± ½
Airflow C3	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	5 ± ½
Airflow C2	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	5 ± ½
Airflow C1	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	5 ± ½
Airstream CZ	½ - 2½	—¼ - + ½	0 - 1/16 (Wee Gee)	5 ± ½
Airstream C6	½ - 2½	—¼ - + ½	0 - 1/16 (Wee Gee)	9½ ± ½
De Soto				
Airflow SG	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	5 ± ½
Airstream SF	½ - 2½	—¼ - + ½	0 - 1/16 (Wee Gee)	9½ ± ½
Dodge 6	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	8½ ± ½
Plymouth				
De Luxe	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	8½ ± ½
6	1 - 3	¼ - ¾	0 - 1/16 (Wee Gee)	8½ ± ½

[a] Caster effect at wheel is obtained by trailing center wheel behind center of King Pin.
[b] Toe in when measured at tire tread, hub height 1/16 in.-⅛ in.

[2] The transverse axis of the sprung mass parallel with the ground line shall mean that the body must be made to assume the same relative position to the steered wheels as when the vehicle is resting on a horizontal plane.
[3] The transverse axis of the sprung mass parallel with the ground line shall mean that the body must be made to assume the same relative position to the steered wheels as when the vehicle is resting on a horizontal plane.

Car	Caster, Deg.	Camber, Deg	Toe In, In.	King Pin Inclination, Deg.
Ford	7 Loaded	2	3/16 - 1/4	7
Graham				
72, 75	4½ - 5½	1	1/8 - 3/16	7½
73	4½ - 5½	1	1/8 - 3/16	7½
74	6 - 7	1	1/8 - 3/16	7½
Hudson				
G-GU Ter.	3¼ - 3¾	1 - 1½	0 - 1/8	7
GH Ter.	3¼ - 3¾	1 - 1½	0 - 1/8	7
HT,HU-HHU Hudson	4 - 4½	1 - 1½	0 - 1/8	7
Hupp D	1½	1	1/16 - 3/16	7½
O	1½	1¼	1/16 - 3/16	8½
Lincoln	2	1	1/16	7½
Olds 6	1½ - 2¼	1/8 - 1	1/8 - 3/16	4⅝
8	1½ - 2¼	1/8 - 1	1/8 - 3/16	4⅝
Packard				
Std. 8	1	1	0 ± 1/16	9
Sup. 8	1	1	0 ± 1/16	9
12	1	1	0 ± 1/16	9
Packard 120-B	2 ± ½ Top of frame 18 in. ab. ground	1 ± ¼ Top of frame 18 in. above ground	1/16 - 1/8	1½
Pierce	¾ ± ¼	1 ± ½	3/16 ± 1/16	8
Pontiac				
Std. 6	1¼ ± ¼	1½ ± ¼	0 - 1/8 9 in. ab. ground	7⅛
De Luxe 6	3/16 in. Trailing Caster 0 - ¾ deg.	2	0 - 1/16 9 in. ab. ground	7
8	3/16 in. Trailing Caster 0 - ¾ deg.	2	0 - 1/16 9 in. ab. ground.	7
Reo 6A	1½	1	0 - 1/16	8
7S	1½	1	0 - 1/16	8
Studebaker				
1B & 1C	¾	1½	1/16 - 1/8	9½
2A	¾	1½	1/16 - 1/8	9½
1A	1½	1½	1/16 - 1/8	9½
Willys	1 - 2	2	3/32	7½

S. A. E. PRODUCTION MEETING

Hotel Statler, Cleveland, Sept. 18 and 19

Limited to two evening sessions, one of them a dinner, the S.A.E. Production Meeting to be held in Cleveland on Sept. 18 and 19, presents a program which for interest and variety rivals those of meetings which have been planned to take much more time.

William B. Stout, president of the Society, will speak on "Where Do We Go From Here" at the dinner session, Sept. 19. With advance registration for the Third Machine Tool Congress, during which the Production Meeting will be held, already at figures comparable with those of pre-depression years, attendance at the S.A.E. dinner promises to be high.

At the technical session on the evening of Sept. 18, papers will be presented on five different subjects, one "A Quick Trip to the Machine Tool Exhibit."

PRODUCTION PAPERS SESSION

Wednesday, September 18, 8:00 P.M. Ball Room

Chairman—JOSEPH GESCHELIN

Methods of Finishing Transmission Gears—S. O. WHITE, Warner Gear Co.

Application of Induction Heating in Automotive Production—E. L. BAILEY, Dodge Bros. Corp.

Rustproofing and Paint Adherence Technique—E. P. SPRUANCE, American Chemical Paint Co.

Resistance Welding in the Automotive Industry—J. A. WEIGER, P. R. Mallory and Co.

A Quick Trip to the Machine Tool Exhibit—J. R. COX, Thompson Products, Inc.

(Papers will be followed by general discussions)

PRODUCTION DINNER SESSION

Auspices S.A.E. Cleveland Section

Thursday, September 19, 7:00 P.M. Ball Room

Chairman—A. T. COLWELL, Chairman, Cleveland Section

Toastmaster—V. P. RUMELY, Vice-President, S.A.E. Production Activity

Topic—Where Do We Go From Here?

W. B. STOUT, Stout Engineering Laboratories, President of S.A.E.

General Information

RAILROAD FARES—Members of the Society may obtain reduced railroad fare certificate (1 1/3 round trip) from F. F. Selbert, Machine Tool Congress, 1220 Guarantee Title Building, Cleveland.

S.A.E. SESSIONS—These will be held in the evenings only, on Wednesday and Thursday, September 18 and 19, at the Hotel Statler, Cleveland.

S.A.E. PRODUCTION DINNER—Dinner tickets may be secured in advance by both members and non-members of the Society from

A. T. COLWELL, Thompson Products, Inc., Cleveland (Chairman, S.A.E. Cleveland Section).

Society of Automotive Engineers, Detroit Section Office, General Motors Bldg., Detroit.

Society of Automotive Engineers, Inc., 29 West 39th Street, New York City.

No table or place reservations will be made. Dress informal.

Price $1.50

NATIONAL TOOL EXHIBIT—The exhibit will be open daily until 6 P.M. from September 11 to 21. Complete information may be obtained from the Machine Tool Congress office.

Research Committee Regulations Approved

PRESENTED by Chairman Ralph Teetor of the Research Committee at the meeting of the Society's Council held June 16 at White Sulphur Springs, during the Summer Meeting of the Society, the Regulations which appear below were approved by the Council for the guidance of all the research activities of the S.A.E. Besides defining the aims and scope of all research work in progress or to be undertaken, the Regulations provide a clear summary of the procedure to be followed.

Research Committee Regulations
Society of Automotive Engineers

In harmony with the Society's Constitution, By-Laws and Rules, these Research Committee Regulations are provided as a guide for research operations of the Society of Automotive Engineers. It is their aim: to state the objectives of S.A.E. research; to define its scope and organization; to present a concept of its approved activities; and to establish for them a recognized, orderly procedure.

Significant Provisions of the Constitution, By-Laws and Rules:

Reference is made to the following provisions which have a bearing upon the Society's research activities and with which the provisions of these regulations have been made to comply:

Committee Appointments—C 49
Duties of Committees—C 50
Removal of Committee Members—C 51
Publications—C 53
Financial Administration—B 17
Society Delegates—B 29

I. Objectives and Scope:

With full recognition of the significance and utility of research practices, the activities of the Society in this field shall be aimed to encourage and promote the application of the best research principles and methods to automotive problems.

The Society's research shall have for its objective the fostering of cooperative research; the securing from all available sources, through organized effort, of technical information for the use of members, the undertaking to this end of such research projects as appear practical and expedient, and the making of such information more readily available to the industry as a whole or to meet specific needs and requests of industry.

In keeping with these objectives, the main endeavor shall be by well directed cooperative effort to gather and distribute needed facts and technical information on automotive engineering matters of broad and timely interest to industry and the Society's membership.

Such accomplishments and others that are in keeping with the spirit of these regulations shall be achieved mainly through the medium of committee consideration and direction of cooperative research on approved projects sponsored by the Society alone or in cooperation with recognized agencies.

As to scope, S.A.E. research shall embrace only those projects or phases of them that are of broad, fundamental interest in the automotive field and shall avoid such operations as may be better adapted to industrial, commercial or private handling.

Each project undertaken shall first be found to qualify as indicated above. It shall be a condition precedent to the commencement of a project, that its proposal, presented in writing and accepted, shall clearly outline a program with acceptable indications of objectives, scope and adequate financing.

II. Organization of Committees:

Any and all committees within the Research organization, and representatives of such committees working cooperatively with other organizations, shall be responsible to the Research Committee.

1. *Research Committee*

(a) The Research Committee shall be composed of interested and qualified members of the Society appointed by the Council for each administrative year, and shall include the Chairmen of the Research Subcommittees. Where the interests of a Professional Activity are not in fact represented on the Committee, the Council, with the advice of the Research Committee Chairman, shall appoint a representative to serve for that Activity.

(b) The Research Committee shall act under the direction of a Chairman, designated by the President, who shall be Chairman also of the Research Executive Committee and ex-officio member of each subcommittee.

2. *Research Executive Committee*

The Research Executive Committee, under the chairmanship of the Research Committee Chairman, shall be composed of the Chairmen of the Research Subcommittees.

3. *Research Subcommittees*

The Research Subcommittees (designated by the Council) shall be composed preferably of members of the Society qualified to undertake such specific committee assignments. Subcommittee members shall be appointed by the Council and Chairmen shall be designated by the President.

4. *Research Divisions*

Subject to the approval of the Council, Subcommittees may be resolved further into such Divisions as occasion may demand or the Research Committee may deem expedient. Members of Divisions shall be appointed by the Council and Chairmen shall be designated by the President with the cooperation of the respective Subcommittee or its Chairman.

5. *Special Committees*

As occasion may demand, the Council may appoint Special Committees to serve under a Chairman designated by the President, with approval of the Council, in an advisory capacity to the Research Committee, Research Executive Com-

mittee, or the Research Subcommittees. In like manner Special Committees may be appointed to function jointly or in cooperation with other organizations.

III. Committee Duties and Responsibilities:

1. *Research Committee*

(a) The Research Committee shall have jurisdiction, subject to the direction and approval of the Council, over the research activities of the Society. Such activities may include work within the Society or in cooperation with other organizations.

(b) The Research Committee shall have authority, subject to the direction and approval of the Council, on matters of policy, finance and execution.

(c) The Research Committee shall guide and direct the efforts of its Subcommittees and Special Committees.

(d) The Research Committee may be charged with such special investigations as the Council may direct.

2. *Research Executive Committee*

(a) The Research Executive Committee shall advise the Research Committee and shall act as an alternate to it on such subjects as are delegated by the Research Committee, its actions being subject to review by the Research Committee.

(b) The Research Executive Committee shall function as an interim committee for the Research Committee on details of policy, execution and finance, all matters requiring Council action being first referred to the Research Committee.

(c) The Research Executive Committee, as occasion may arise, shall investigate and report to the Research Committee on the propriety and practicability of new research projects from the viewpoints of scope, program, necessity, timeliness, extent of demand, service to be rendered and adequacy and equity of financing arrangements.

(d) The Research Executive Committee at least semi-annually, shall investigate and report to the Research Committee on the advisability of curtailing, suspending, or relinquishing to another organization any existing research project.

(e) The Research Executive Committee may, in the absence of any special committee designated for that purpose, act as a liaison group between the Research Committee and similar divisions of other Societies.

3. *Research Subcommittees*

Research Subcommittees, subject to approval of the Council, shall be charged by the Research Committee with the conduct of specific projects falling within a prescribed scope.

4. *Research Divisions or Sections*

Research Divisions shall be given such detailed assignments within the scope of the respective Subcommittee, as that Subcommittee may designate.

5. *Special Committees*

Special Committees shall serve as authorized by the Council, under direction of the Research Committee.

IV. Procedure:

(a) Meetings of the Research Committee shall be held in conjunction with each Annual and Semi-Annual Meeting of the Society. Interim meetings may be held at the call of the Chairman or by the direction of the Council.

(b) Meetings of the Research Executive Committee, Research Subcommittees, Divisions and Special Committees may be called at the request of the respective Chairmen of the subdivisions or the Research Committee Chairman or on due notice in writing by the Research Manager or the Secretary of the Society.

(c) The Chairman of a Research Subcommittee may delegate in writing any member of the Subcommittee to represent him at a stated meeting of the Research Committee and/or the Research Executive Committee.

(d) In the absence of the Chairman at the meeting of the Research Committee or any of its subdivisions, the meeting concerned shall select a member to fill the Chair.

(e) One-fourth of the membership, with a minimum of three members, shall constitute a quorum at meetings of the Research Committee, the Research Executive Committee, Research Subcommittees, Divisions and Special Committees.

(f) The order of business at Committee Meetings shall be:

Attendance Registration
Approval of Minutes
Subject in Progress
Proposal of New Subjects
Miscellaneous Business
Time and Place of Next Meeting

(g) The Research Manager or a representative of the Research Department of the Society shall record the general minutes, including the names of those in attendance at or absent from all Committee, Subcommittee and Division Meetings. In the absence of such representative a suitable record of the proceedings of such meeting or of members thereof shall be made by, or by direction of, the Chairman or chairman pro tem. of the meeting.

(h) After completion of work on any subjects, any official papers or correspondence relating thereto shall be transmitted to the Research Manager by Chairmen or Committee Members.

(i) No matter resulting from the work of research committees shall be issued for publication unless or until such release is approved by the Research Committee, Research Executive Committee, the Secretary or Research Manager in writing.

V. Revisions:

Revisions of, or amendments to, these regulations may be made by the Council.

Sept. 27, 1934
Revised: Jan. 7, 1935

Constitutional Amendment Voted

In a mail ballot sent to members of the Society, July 12, 1935, a proposed addition to the S.A.E. Constitution governing admission of past-presidents of the Society to life membership was adopted by a vote of 802 to 76 valid ballots.

The amendment reads as follows:

C-24a—Past-Presidents of the Society shall be admitted to Life Membership without payment of a Life Membership fee at the end of their second year's service on the Council as Past-President.

The tellers of election—George A. Round, A. E. Becker and F. K. Glynn—certified the votes as follows:

Ballots favoring adoption	**802**
Ballots not favoring adoption	**76**

Credo...

"I believe the S.A.E. is the most efficient society in the automotive industry. The standardization it has accomplished has greatly simplified our problems. It is a great medium for the interchange of constructive ideas and offers tremendous advantages to the younger engineers."

—GUY W. VAUGHAN

Guy W. Vaughan . . . president, Curtiss-Wright Corp. . . . engineer . . . executive . . . in the automobile industry for many years . . . entered aviation as quality manager on American Hissos during the War . . . a member of the S.A.E. since 1908.

THIS PAGE IS PRINTED FOR THE INSPIRATION OF OUR PRESENT MEMBERS—FOR PRESENTATION TO PROSPECTIVE MEMBERS

New Members Qualified

These applicants who have qualified for admission to the Society have been welcomed into membership between July 10, 1935, and Aug. 10, 1935.

The various grades of membership are indicated by: (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate Member; (SM) Service Member; (FM) Foreign Member.

Amos, James Morse (A) assistant superintendent, transportation, Overnight Motor Transportation Co., 101 South Fremont Avenue, Baltimore, Md.; (mail) 1248 East North Avenue.

Anderssen, Frithjof (A) sales manager, Aktieselskabet Bil, Trondhjen, Norway; (mail) Thomas Angels gt. 3, Trondhjem, Norway.

Bandi, Alfred (A) director, Scintilla of Canada, Ltd., 3483 Park Avenue, Montreal, Quebec, Canada.

Cumming, A. A. (J) sales engineer, Canadian National Carbon Co. Ltd., 805 Davenport Road, Toronto, Ontario, Canada.

Edge, Stanley Howard (FM) designer, Clayton-Dewandre Co., Ltd., Lincoln, England; (mail) "Frankley" Hawthorn Road, Cherry Willingham.

Edworthy, Thomas P. (A) manager, Prairie Cities Oil Co., Ltd., Winnipeg, Manitoba, Canada.

Garberg, James (M) engineer, plant and equipment, Minnesota State Highway Department, 1246 University Avenue, St. Paul, Minn.; (mail) 5032 Vincent Avenue South, Minneapolis.

Gardner, C. V. (M) engineer, Hydraulic Brake Co., 84 West Hancock Avenue, Detroit.

Gibbs, E. R. (A) assistant manager, Socony-Vacuum Oil Co., 59 East Van Buren Street, Chicago.

Gould, Jack (A) assistant maintenance engineer, Saskatchewan Power Commission, Regina, Sask., Canada.

Haferl, Rolf C. (M) design engineer, General Motors Truck Corp., Pontiac, Mich.; (mail) 30 Williams Street.

Hodgson, Alfred (A) service manager, Trinidad Agencies, Ltd., 40 South Quay, Port of Spain, Trinidad, British West Indies; (mail) 30 Mucurapo Road.

Johnson, Glenn D. (M) chief engineer, MacClatchie Mfg. Co., Box 651, Compton, Calif.

Joyce, Reginald (J) mechanical engineer, Consumers' Research, Inc., Bowerstown, Washington, N. J.; (mail) 15 Gardner Court.

Laurie, Gavin W. (M) manager, automotive transportation, Atlantic Refining Co., 260 South Broad Street, Philadelphia.

Levine, Sam (A) president, Hackensack Auto Parts, Inc., 54 State Street, Hackensack, N. J.

McConnell, Charles Wm. (J) research engineer, Hyvis Oils, Inc., Warren, Pa.; (mail) 11 Jefferson Avenue.

Noell, Milton James (J) student, Oil Well Supply Co., Oil City, Pa.; (mail) Y. M. C. A.

Parks, John L. (M) shop superintendent, Horton Motor Lines, 1410 North Monroe Street, Baltimore, Md.

Riesing, Ellwood F. (A) sales engineer, Firestone Tire & Rubber Co., Akron, Ohio.

Sayre, John S. (A) vice-president, branch manager, Highway Trailer Co., 2001 South Wabash Avenue, Chicago.

Schramm, Milton E. (J) motor testing engineer, Shell Petroleum Corp., East Chicago, Ind.; (mail) 4241 North Paulina Street, Chicago.

Schroeter, Fred J., Jr. (J) engineer, Ethyl Gasoline Corp., 716 East Milwaukee Avenue, Detroit; (mail) 2051 West Grand Boulevard.

Seese, Melvin Edward (J) tool grinder, Studebaker Corp., South Bend, Ind.; (mail) 1024 North Allen Street.

Selley, Charles Henry (FM) deputy manager, motor department, Shell-Mex & B. P. Ltd., P. O. Box 148, Victoria Embankment, London W. C. 2, England.

Vinson, Robert W. (A) mechanic, Wentworth & Irwin, Inc., 1005 West Burnside Street, Portland, Ore.

Whipple, Leon Vincent (J) tool designer, Trico Products Corp., 817 Washington Street, Buffalo, N. Y.; (mail) 414 Linden Avenue, East Aurora, N. Y.

Whittelsey, Ralph Kinney (A) representative, Bendix-Westinghouse Co., Securities Building, Seattle, Wash.

Wilson, W. E. (J) Bower Roller Bearing Co., Detroit; (mail) 1941 Wellesley Avenue.

Wirshing, Herbert (A) western representative, Waukesha Motor Co., Waukesha, Wis.; (mail) 150 Lake Street, Oakland, Calif.

Applications Received

The applications for membership received between July 15, 1935, and Aug. 15, 1935, are listed herewith. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

Bassi, Andrew B., automotive engineer, Socony-Vacuum Oil Co. Inc., Chicago, Ill.

Bennetch, David S., Sheridan, Pa.

Brandt, Frederick C., designer, Hudson Motor Co., Detroit, Mich.

Carper, William F., division lubricating engineer, General Petroleum Co., Portland, Ore.

Farmer, Harold Oatridge, chief engineer, Petters Ltd., Yeovil, England.

Farnum, Samuel Barnard, Jr., junior engineer, Standard Oil Co. of N. J., New York City.

Fifield, George A., assistant manager, Sparta Foundry Co., Sparta, Mich.

Hess, Robert A., engineer, Standard Oil Co. of N. J., New York City.

Hutton, Charles Gordon, engineer, Heaps Engineering Co. Ltd., New Westminster, B. C.

Jacobs, Charles John, managing director, John McGrath Ltd., Sydney, N. S. W., Australia.

MacDonald, Gilmour C., supervising mechanic, Soil Conservation Service, Bethany, Mo.

Pointer, Robert William, manager truck equipment dept., Beall Pipe & Tank Corp., Portland, Ore.

Smallman, Frank E., Mahwah, N. J.

Smith-Pert, George James, assistant designer, Fairey Aviation Co. Ltd., Hayes, Middlesex, England.

Shepard, David A., European representative, Standard Oil Development Co., New York City.

Sherratt, Ernest, designer, E. R. Foden & Son, Sun Works, Sandbach, Cheshire, England.

Suda, Jerry, 20-46 29th St., Astoria, L. I., N. Y.

Tett, Hugh Charles, manager, Technical Dept., Anglo-American Oil Co. Ltd., London.

Tout, Virgil H., district manager, Raybestos Division, Denver, Colo.

Van Drongelen, Victor John, 3637 14th St. West, Seattle, Wash.

Wilson, Arthur H., sales engineer, Peter Pirsch & Sons Co., Kenosha, Wis.

Young, James Godfrey, general manager, Sanderson Bros. & Newbold Co. Ltd., Sheffield, England.

"Sky Hooks" for Automobiles

By W. S. James, H. E. Churchill and F. E. Ullery
Studebaker Corp. of America

IN this paper the authors present some experimental results obtained by using the analysis outlined by Prof. James J. Guest before the Institution of Automobile Engineers, in 1926. To make the experimental work more understandable, they present the essential points of Professor Guest's analysis.

Professor Guest begins his analysis of the movements of a car body with the simplest set of conditions and presents a graphical as well as an algebraic solution. He then includes one additional factor after another in his analysis until the principal factors in car suspension are included. After all factors are considered, the essential structure of the simple analysis is retained.

The authors' efforts at the experimental determination of the moment of inertia of passenger cars were started in January, 1932, on Sir Charles Dennistoun Burney's "tear-drop" design with which he visited leading American manufacturers. The first of the series of experiments was made by jouncing the car alternately about the front and rear axles. The front springs were blocked rigidly and the car jounced on the rear springs to obtain the period of the horizontal pendulum thus created. A further series of tests was made on a small laboratory set-up.

The foregoing and other check tests are described in detail and the results commented upon. The development of the formula used in calculating the moments of inertia obtained experimentally with a horizontal compound pendulum is given, further to clarify the method described.

In conclusion, the authors state that it is not possible to draw any definite conclusions from the work described, but they feel that the analysis presented by Professor Guest has made it possible to visualize clearly many of the problems connected with vehicle springing.

THE basic theory of vehicle spring-suspension has been outlined quite frequently; however, there is a rather meager amount of experimental data on actual cars which can be used in connection with such mathematical analysis. In this paper the authors wish to present some experimental results obtained by using the analysis outlined very clearly by Prof. James J. Guest before the Institution of Automobile Engineers, in 1926. To make the experimental work more understandable, an attempt will be made to present the essential points of Professor Guest's analysis. For further details, those interested should consult the original paper.

Professor Guest begins his analysis of the movements of a car body with the simplest set of conditions, and presents a graphical as well as an algebraic solution. He then includes one additional factor after another in his analysis until the principal factors in car suspension are included. After all factors are considered, the essential structure of the simple analysis is retained.

The rotational motion of the sprung mass of a car about its center of gravity, G, will be the same as the rotation of a long thin cylinder equal in weight to the sprung mass and having a radius, K, equal to the radius of gyration of the sprung mass. (Fig. 1). The thin cylinder can be replaced by two weights, M, each one-half the sprung weight, provided their centers of gravity are a distance $2K$ from each other. (Fig. 2). This weight system can be replaced by an infinite number of systems, provided the masses and their distances from the center of gravity are as shown in Fig. 3. The graphical method used by Professor Guest for showing the possible combinations of p and q is given in Fig. 4. A hyperbola $pq = k^2$ is constructed, using ordinates through the center of gravity and the value of k as the radius of gyration. A 45-deg. line through the center of gravity then serves to transfer the value of p to the horizontal line. All conjugate points p and q are easily obtained by this method.

Professor Guest then considers the motion of a stiff weightless bar resting on two springs of stiffness λ and μ, in pounds per inch. If such a bar, as shown in Fig. 5, is acted on by a vertical force at the point C, it will move down parallel to its original position. This point Professor Guest calls the spring center, and its relation to the points at which the springs act and the spring rate is $\lambda\ AC = \mu\ BC$. When a force is applied to the weightless bar at some other point, such as P, the final position of the bar if extended will intersect the line of the original position at some point Q. If the same force were originally applied at Q, the final position of the bar would be such that its extended line would intersect the original position at the point P. These two points are con-

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 19, 1935.]

jugate, and their relation to the spring rates and the points of spring support can be expressed as $PC \times QC = h^2$, where h is a constant and equal to:

$$\frac{\sqrt{\frac{\lambda}{\mu}}}{1 + \frac{\lambda}{\mu}} \cdot AB$$

This is a hyperbolic relation which Professor Guest shows graphically, as in Fig. 6.

Having shown that the weight system and the spring system can be graphically presented as hyperbolic relations, Professor Guest combines the two, as in Fig. 7.

In Fig. 7, the spring system is represented as in Fig. 6 and the mass system is superimposed on it by locating the point representing the center of gravity, G, on the 45-deg. line through the spring center, C. The intersection of the two hyperbolas at the point R locates the two conjugate points P and Q. The motion of the sprung mass of a car will be that of independent rotation about these two points P and Q as centers; the weight of the car being considered as divided into two parts, related as shown in Fig. 3. Following the nomenclature of Fig. 5, the period of vibration of the point P rotating about Q is:

$$T_1 = \frac{2\pi}{a+b}\sqrt{\frac{M}{g}\cdot\frac{q}{p}\left[\frac{(p+b)^2}{\lambda} + \frac{(a-p)^2}{\mu}\right]}$$

and that of the point Q about P is:

$$T_2 = \frac{2\pi}{a+b}\sqrt{\frac{M}{g}\cdot\frac{p}{q}\left[\frac{(q-b)^2}{\lambda} + \frac{(a+q)^2}{\mu}\right]}$$

In discussing the motion of the body about the points P and Q, Professor Guest states: "These simple harmonic vibrations are completely independent of one another. If the car has one of these vibrations only, it may be considered to rock on a fixed axis at P (or Q) and its vibration calculated as if the axis were actually fixed. If the two modes of vibration exist simultaneously, each vibration will pursue its course uninfluenced by the other, and the movement of any point of the car can be found by the combination of the movements. Thus, with definite conditions as to the start of the motion, the position, velocity, and acceleration, of any point of the car at any time can be found. The vibrations of the car body are usually generated by forces transmitted through the springs at A and B; accordingly, both modes of oscillation are usually impressed simultaneously upon the car. The motion of any point U will thus, in general, consist of the sum of two simple harmonic motions, and its motion will be by no means simple, particularly as the periods T_1 and T_2 will usually have no simple fractional relationship such as would cause the motion to repeat after a short number of simple periods. The points P and Q themselves will, however, only receive vertical motion; each from the angular oscillation about the other."

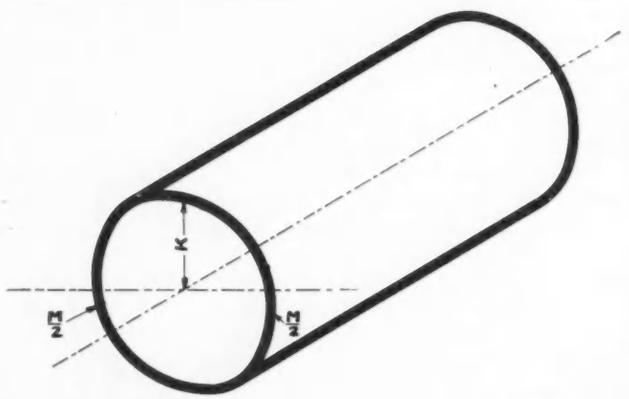


Fig. 1—Rotational Motion of the Sprung Mass of a Car Compared with That of a Long Thin Cylinder

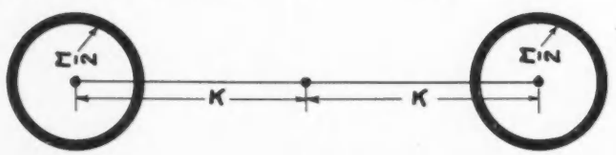


Fig. 2—Replacement of the Cylinder Shown in Fig. 1 by Two Weights

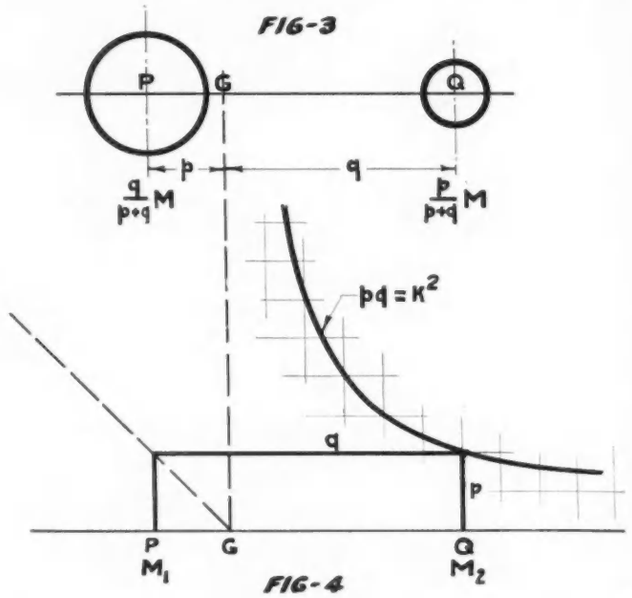


Fig. 3 (*above*)—Replacement of the Weight System of Fig. 2 by an Infinite Number of Systems

Fig. 4 (*below*)—Graphical Method Used by Professor Guest for Showing the Possible Combinations of p *and* q

This rather simple analysis of the motions of the car body assumes that the center of gravity, G, is in the same horizontal plane as the axles. This, however, is never the case. Professor Guest then shows that there is some point in the vertical line through the center of gravity which has vertical motion only. This point lies above the plane of the wheel centers and below the center of gravity. The vertical location of this point is dependent upon the ratio of the sprung mass to the effective unsprung mass, considering these two masses as balancing each other when rotated about a horizontal axis through these points; see Fig. 8. The conjugate centers of oscillation will, therefore, lie in a horizontal line through this point, designated by Professor Guest as at a distance l below the height of the center of gravity.

Professor Guest then analyzes the angular movements of the car with the center of gravity being above the line through the conjugate centers P and Q, showing that the original relations still hold if k^2 is replaced by $(k^2 + l^2)$ and that the spring constant, h^2, is replaced by $h^2 - lu$, in which u is the distance

the center of gravity of the sprung mass must be raised to unload the springs completely from the fully loaded position.

This modification of the simpler relation changes the graphical solutions, as indicated in Fig. 7, the semicircle of radius $(a + b)/2$ being replaced by the slightly shorter radius

$$\sqrt{[(a+b)^2/(4-lu)]}.$$

The next major factor which Professor Guest introduces is the effect of the spring action of the tires on the mass between the tires and the main car-springs. After a careful analysis, he shows that these factors can be allowed for by considering the stiffness of the car springs λ and μ to be compounded with springs of stiffnesses $(\lambda_2 - m_1\omega_1^2)$ and $(\mu_2 - m_2\omega_1^2)$ giving in place of λ the expression:

$$\frac{\lambda(\lambda_2 - m_1\omega_1^2)}{\lambda + (\lambda_2 - m_1\omega_1^2)}$$

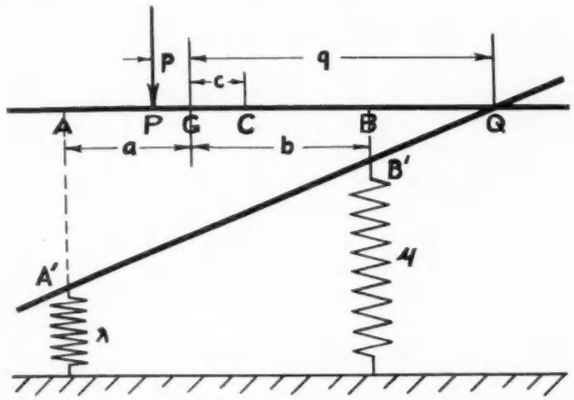


Fig. 5—Motion of a Stiff Weightless Bar When Acted upon by a Vertical Force at the Point *C*

and for μ the expression:

$$\frac{\mu(\mu_2 - m_2\omega_1)^2}{\mu + (\mu_2 - m_2\omega_1^2)}$$

where λ_2 and μ_2 are the respective tire-spring rates, m_1 and m_2 the unsprung masses between the tire and the sprung mass, and ω the time, t, per cycle of oscillation divided by 2π, or $\omega = t/2\pi$.

To sum up in Professor Guest's words, he has stated: "With a simplified view of the car, the particular aspect from which the mass system is regarded and a theorem developed concerning the elastic system enable a clear idea to be obtained of the nature—rocking oscillations about a pair of axes—of the simple primary motions which underlie the main complex vibration, and certain properties and relationships of the system to be established. After considering friction, and rolling unsymmetrical springing, the author has shown that, with certain alterations in the values of a pair of quantities, the conclusions reached apply to a car upon simple springs, and are, thus corrected, practically adequate. Finally, he has shown that the effect of the axle masses on the primary vibrations can be allowed for in a simple manner, and that, in the pair of secondary vibrations which they introduce, the car body rocks about a pair of axes, the positions of which are readily found."

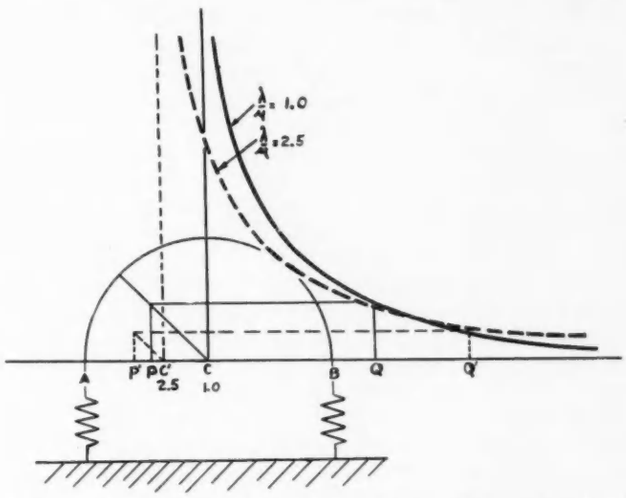


Fig. 6—Hyperbolic Relation of the Points *P* and *Q* to the Spring Rates and the Points of Spring Support

It is evident that, to check this analysis against actual movements of the car, it is necessary to know the:

(1) Radius of gyration of the sprung mass, k
(2) Front and rear spring rates μ and λ
(3) Vertical and horizontal locations of the center of gravity
(4) Spring rate of the tires
(5) Unsprung mass
(6) Moment of inertia of the wheels and tires
(7) Frictional or damping forces acting on the springs and tires

Most of these factors can be determined by well-known methods. Some, however, are not obtained in the regular practice, particularly k, the radius of gyration of the sprung mass.

Our efforts in the experimental determination of the moment of inertia of passenger cars were started in January, 1932,

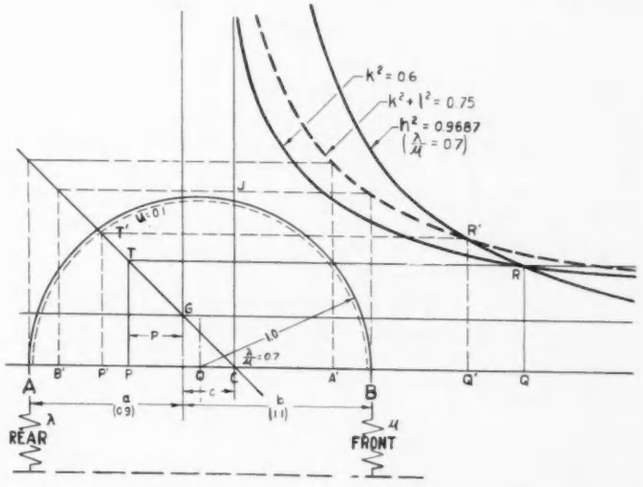


Fig. 7—Combination of the Weight System and the Spring System

The spring system is represented as in Fig. 6 and the mass system is superimposed on it by locating the point representing the center of gravity, *G*, on the 45-deg. line through the spring center, *C*.

on Sir Charles Dennistoun Burney's "tear-drop" design with which he visited leading American manufacturers. This car, well streamlined, independently suspended front and rear, and with the engine mounted in the rear, had exceptionally good riding qualities. Sir Dennistoun designed this car to have its radius of gyration, squared, equal to the product of the distance from the front-wheel centers to the center of gravity and the distance from the center of gravity to the rear wheels. His theory, well discussed in his paper[1], is that a car built on this principle will have the best riding qualities obtainable as far as they can be effected by design.

The first of the series of experiments was made by jouncing the car alternately about the front and rear axles, following the procedure of experimental determination outlined in a paper[2] by H. S. Rowell. The front springs were blocked rigidly and the car was jounced on the rear springs to obtain the period of the horizontal pendulum thus created. This procedure was reversed for determining the period of the car with the rear springs blocked rigidly.

Calculations for k^2 from the data thus obtained gave us a k^2/ab ratio very much greater than Sir Dennistoun's previously calculated ratio of unity. The errors of this determination were attributed to our inability to obtain an accurate period of the pendulum due to friction and tire deflection. Friction in the suspension system of the car necessitated the application of considerable force to maintain oscillation. The variation of force applied and of phase of application disturbed the simple harmonic motion necessary to obtain a correct period of oscillation, probably because the disturbing force was not applied at or close enough to the oscillating centers P and Q. Removing the action in the tires by anchoring the axle improved results but little.

Swinging the car body as a pendulum called for a rigid point of support which was not available. A horizontal cradle was constructed of I-beams, which was supported at one end by shafting and heavy A-frames and at the other by leaf springs as shown diagrammatically in Fig. 9. The tires were removed and the wheels mounted in castings fitting the rims.

[1] See S.A.E. JOURNAL, February, 1932, p. 57; "The Development of a Rear-Engined Streamlined Car."

[2] See *Proceedings of the Institution of Automobile Engineers*, part 2, 1922-1923, pp. 455 and 565; "Principles of Spring Suspenson", by H. S. Rowell.

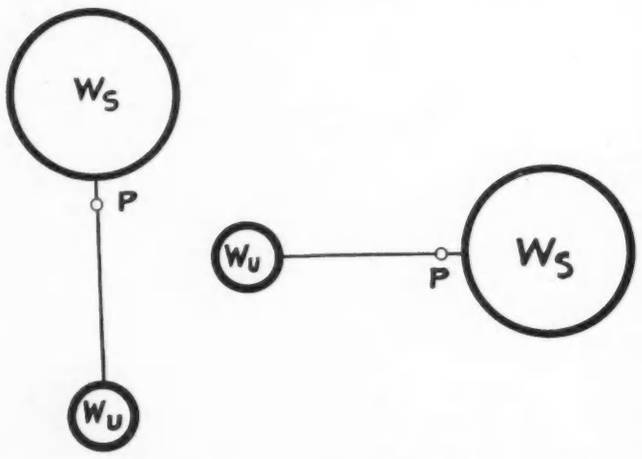


Fig. 8—Illustration of How the Conjugate Centers of Oscillation Lie in a Horizontal Line Through a Point at a Distance l Below the Height of the Center of Gravity

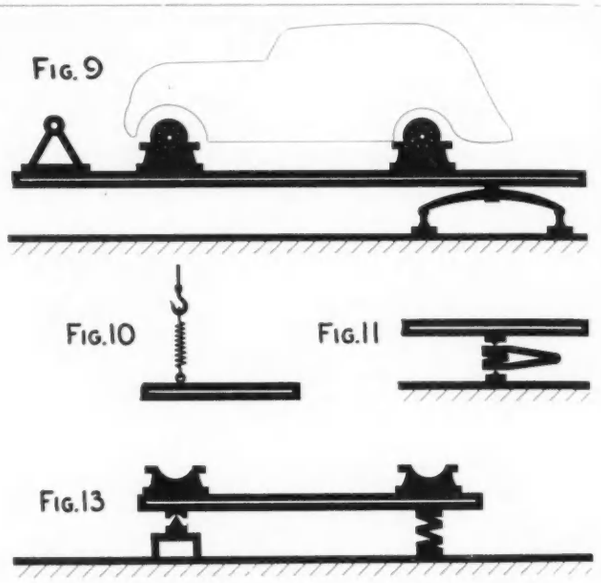


Fig. 9—Horizontal Cradle Constructed of I-Beams

Fig. 10—Cradle Supported on Tension Coil-Springs

Fig. 11—Springs Mounted To Act Through Knife Edges on the Horizontal Cradle

Fig. 13—Construction Believed by the Authors To Be the Most Satisfactory and Least Expensive

The springs were then blocked in their normal position and the cradle and car swung by hand as close to its natural frequency as possible. The car was then removed and the cradle supported on tension coil-springs (Fig. 10) and its moment of inertia determined. From these two periods and the previously determined locations of the center of gravity of the car and cradle, the moment of inertia of the car was computed. The results obtained from this horizontal pendulum employing high-rate leaf-springs gave a k^2, for Sir Dennistoun's car, equal to 38.5 lb-ft.^{2} or a k^2/ab ratio of 0.91, which caused us to assume that an error of 10 per cent still existed.

It was found at once that the friction in the leaf springs made it difficult to determine the true period. In addition, we did not believe the results. To check the results, several die blocks were obtained of a total weight about equal to the car and their moment of inertia was determined experimentally and compared with computed values. A battery of coil springs of approximately four times higher rate than the leaf springs was installed, and with this combination a still greater error was obtained. However, it was observed that the pendulum would oscillate at its natural frequency, without external force applied, for a much longer period than had previously been observed with the lower-rate leaf-springs. This permitted us to obtain a very definite natural frequency of the system; hence, the further use of leaf springs was abandoned. Data were then taken on the cradle only with high-rate and with low-rate coil-springs. The comparative results showed the experimental determination of the moment of inertia of the cradle only, with low-rate springs, to be only 2.2 per cent smaller than the calculated moment of inertia. It was concluded from these series of tests (Table 1) that low-rate springs gave the best results because the period of the supporting springs was much lower than the natural frequency of the cradle I-beams.

Table 1—Comparison of Experimental and Calculated Moments of Inertia with Various Supporting Springs

	Total Weight, Lb.	Spring Rate, Lb. per In.	Moment of Inertia, Lb-Ft². Experimental	Moment of Inertia, Lb-Ft². Calculated	Error, Per Cent
(1) Die Blocks with Leaf Springs...	6,715	638	20,400	19,500	+ 4.5
(2) Die Blocks with Coil Springs...	6,715	2,215	22,100	19,500	+13.0
(3) Horizontal Cradle; Low-Rate Coil-Springs.....	1,665	123	8,780	8,980	− 2.2
(4) Horizontal Cradle; High-Rate Coil-Springs.....	1,665	559	6,390	6,710	−10.0

The large errors in the results obtained with full-scale tests gave rise to the question as to the correctness of the method of using a horizontal pendulum. To answer this question, a further series of tests was made on a small laboratory set-up further to check the accuracy of the larger cradle. This apparatus consisted of a test stand on which was mounted a frictionless pivot for a horizontal rod. The opposite end of the horizontal rod was supported by a very low-rate coil-spring of practically uniform rate. The upper end of the spring was attached to a fixed support. A sliding ball mounted on the horizontal rod provided a method of shifting the mass of the pendulum. Tests were made with the ball in various positions on the rod and the experimentally determined moments of inertia thus obtained were compared with the calculated moments of inertia. The results of these experiments, which gave a 0.75-per cent average-error and a maximum error of 2 per cent, are shown in Table 2. The development of the relation between the moment of inertia and the spring rate of a horizontal pendulum is given in the Appendix.

The question of effective spring rate and point of action of the springs arose. These questions were answered by calibrating a battery of springs in place and mounting them so that they acted through knife edges on the cradle. (See Fig. 11). While these items were being checked with die blocks, it became evident that the effect of the location of the weights on the cradle was important because of the lack of complete rigidity of the I-beams. A new set of lower-rate coiled-springs was then obtained and the results were rechecked. The results obtained during this work are shown in Fig. 12. It will be noticed that very good results were obtained with the low-rate springs and knife edges between the springs and the I-beam. It will also be noticed that the best results were obtained when the weights were located over the springs and as close to the fixed hinge-point as possible.

Fig. 13 shows the construction which the authors believe to be the most satisfactory and least expensive. In this arrangement no raised hinge support is required, the car weight is located directly over the points of support, and the moment of inertia of the supporting cradle is a minimum. This construction has not been tried, but experience with the use of the arrangements previously described indicates that it would be very satisfactory. In measuring k for various passenger

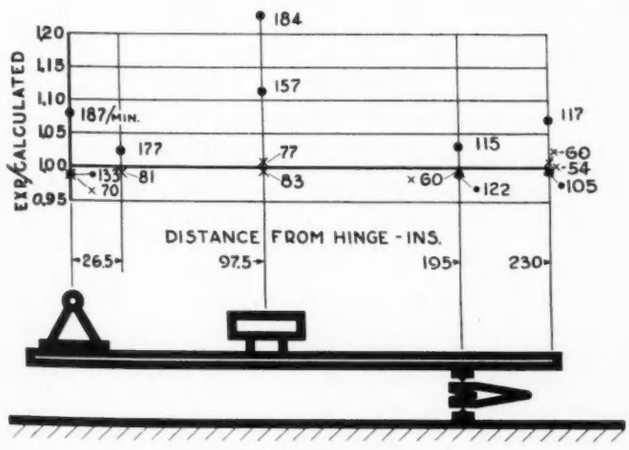


Fig. 12—Check-Up of Results

Stiff springs, each 250 to 300 lb. per in., are designated by the large dots; soft springs, each about 40 lb. per in., on knife edges, by X. The figures beside the points give the frequency of oscillation in cycles per minute.

Table 2—Comparison of Experimental and Calculated Moments of Inertia for Different Locations of the Center of Gravity

Ball Location From Center of Oscillation, In.	Oscillations Per Sec.	Moment of Inertia Experimental	Moment of Inertia Calculated	Error, Per Cent
7	1.145	54.5	54.1	+0.1
8	1.037	66.4	66.6	−0.3
9	0.952	78.9	80.6	−0.2
10	0.868	94.8	95.4	−0.7

loadings, it was necessary to remove the seat cushions, to use weights fastened into place and to remove the gasoline from the tank. The damping action of people sitting on cushions or gasoline swashing about in the gas tank caused the period of oscillation to be indefinite.

The detailed procedure in determining the moment of inertia is, first, to clamp each wheel into its bracket with a hoop to prevent any change in the position of the car during the test. Block all chassis springs in their free carrying position for the particular loading. Install the necessary number of coil springs under the cradle to suspend it properly, and shim between the floor and spring bases to bring the cradle horizontal. At the rear end of the cradle attach a Veeder counter with a short arm. This counter aids in counting the oscillations. Very little effort is required to oscillate the cradle, care being exercised that its vibration is not forced. Five time-readings are taken, each for 100 complete oscillations. These time-readings seldom vary more than 0.15 sec. in 80 to 100 sec. for the 100 complete oscillations. The amplitude at the counter is about ½ in.

The coil springs are calibrated for each passenger loading by a load-deflection test. The load is applied at the rear end of the cradle with accurate weights. The arc of deflection is measured with dial extension gages at right angles to a line from the hinge to the gage point. The angular spring-rate in inch pounds per radian is:

$$\frac{\text{Moment (In-Lb.)}}{\text{Arc (Radians)}} = \frac{\text{Load (Lb.)} \times \text{Horizontal Distance from Hinge to Load (In.)}}{\dfrac{\text{Arc (In.)}}{\text{Radial Distance from Hinge to Gage}}}$$

This figure, for the total car, is usually about 12,000,000 in-lb. per radian.

The period and the spring constant determined, the moment of inertia of the car and cradle about the hinge is given by the formula:

$$\frac{12 \times 32.16(g) \times (\text{Period, Sec.})^2 \times \text{Angular Rate (In-Lb. per radian)}}{4\pi^2}$$

The derivation of this formula is given in the Appendix. This numerical value varies from about 75,000,000 to 110,000,000 lb-in.^{2} depending on the car and the loading.

This procedure is followed for each loading; road less gas, road, road plus driver, road plus three passengers, and road plus five passengers. Each "passenger" consists of three standard 50-lb. weights. The location of the passengers, as well as the gas-tank location, is measured with respect to the rear-wheel center-line.

The sprung mass of the car is removed from the cradle. The unsprung mass consists of wheels, brakes, axles (or one-half of swinging arms) and one front and one rear spring in position. These parts are all clamped in proper position. The cradle is properly suspended, and the period and spring constant determined as before. This gives the moment of inertia of the unsprung mass less tires, and the cradle parts, about the hinge. This figure is about 39,000,000 lb-in.^{2}, of which about 28,000,000 lb-in.^{2} is the moment of inertia of the cradle. The moment of inertia of the cradle and unsprung mass subtracted from the moment of inertia of the complete car and cradle gives the moment of inertia of the sprung mass about the hinge, which is of a magnitude of 38,000,000 to 59,000,000 lb-in.^{2}, depending on the loading.

The unsprung parts are removed from the cradle, are weighed and the centers of gravity of the front and of the rear assemblies are determined. From these data the total unsprung weight and the center of gravity, with respect to the wheel centers, are calculated. Having the center of gravity and weight of the car complete and that of the unsprung mass, the weight and center of gravity of the sprung mass are readily determined. The horizontal center of gravity of the sprung mass usually shifts forward for light loadings and backward for heavy loadings from that of the complete car. The vertical center of gravity of the sprung mass is 2 to 3 in. above that of the complete car.

The location of the front-wheel center from the hinge was measured with the car on the cradle. The front-wheel center usually is about 50 in. horizontally from the hinge and, vertically, about 6 in. below it. Thus, the distance of the center of gravity of the sprung mass from the hinge is known. From the moment of inertia at the hinge, subtract the product of the unsprung mass times the square of its distance from the hinge. This product is a magnitude of from 32,000,000 to 40,000,000. The moment of inertia of the sprung mass at its center of gravity for a moderate-sized car ranges from 6,600,000 to 8,000,000 lb-in.^{2} from "road less gas" to "five passenger loading".

When the car was on the cradle, the locations of the passengers and the gasoline were measured. Thus their effect can be computed directly. These effects, subtracted from the respective experimentally determined moment of inertia, give five figures for "road less gas" loading which should be equal. This serves as a partial check.

When the car is still apart, the heavier rotating masses are swung as compound pendulums on a knife edge and their moment of inertia determined. Usually one wheel, tire and tube assembly, and one front and one rear brake drum, are tested in this manner. These parts have a small effect on the height of the horizontal plane of the points of oscillation.

The spring-rate determination is merely a deflection-load test. The sprung mass is deflected with respect to the wheel center and the change of tire load measured. In our work the car is placed on four platform-scales, permanently set in the floor, one under each wheel, and the car is weighed. Fastened to the floor around the car is a steel structure with adjustable cross-members. On these cross-members are long power-screws with hand-wheels. To the end of each frame horn, one of these screws is attached through a swivel nut and bracket. On each fender, directly over the wheel, a reference mark is scribed. The entire sprung mass may be

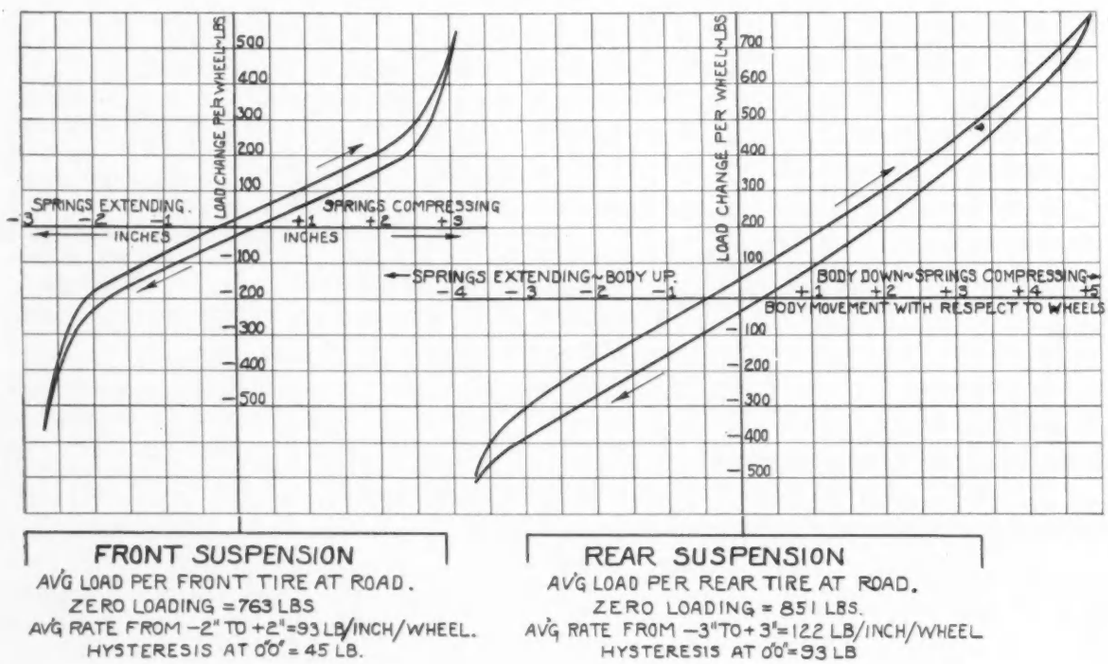


Fig. 14—Representative Load-Deflection Curves of Front and Rear-Wheel Spring-Rates, Showing the Hysteresis Loop

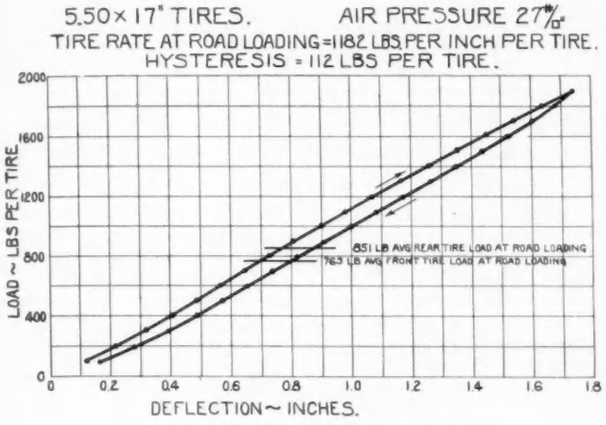

5.50×17" TIRES. AIR PRESSURE 27#/□"
TIRE RATE AT ROAD LOADING = 1182 LBS. PER INCH PER TIRE.
HYSTERESIS = 112 LBS PER TIRE.
LOAD ~ LBS PER TIRE
2000
1600
1200
800
400
0
851 LB AVG REAR TIRE LOAD AT ROAD LOADING
763 LB AVG FRONT TIRE LOAD AT ROAD LOADING
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8
DEFLECTION ~ INCHES.

Fig. 15—Representative Tire-Rate Curves

raised or lowered by the screws as desired. Usually, only one end of the car is changed. The vertical deflection being small, compared to the length of the car, the angular error is immaterial. Leaving one end of the car at the normal position, the other end is raised until there is only about 150 lb. on each tire. The scale readings and the heights of the fender mark from the scale platforms are measured as the sprung mass is lowered by increments of ½ to ¾ in. until the spring bumpers are well compressed. The same procedure is followed as the springs are returned to their extended position.

These results include the tire deflections, which are corrected from the tire load-deflection curves to give the net deflection between the sprung mass and the wheel center. These results are plotted about the normal carrying position as zero. The average slope indicates the spring rate. Usually, this is taken over a range from about 1½-in. extension to 2-in. compression for the front suspension; for the rear, about 3-in. extension to 3-in. compression. The spring rates of independent front suspensions range from about 90 lb. per in. to 150 lb. per in. per wheel. Conventional front suspensions may range from 140 lb. per in. to 350 lb. per in. per wheel. For rear suspensions, from 120 to 150 lb. per in. per wheel. The hysteresis loop indicates the total static friction in the suspension at the normal loading. For greased leaf-springs, this friction varies from 70 lb. to 150 lb. per wheel. For independent coil-spring suspensions, it is from 40 to 60 lb. per wheel. A representative curve of front and rear-wheel spring-rates, showing the hysteresis loop, is given in Fig. 14.

The tire-rate determination is a simple load-deflection test at the recommended air pressure. In our tests, a stiff beam about 20 in. long with wheel hubs welded to each end is used with a wheel and tire assembly bolted to each end. This complete assembly is mounted on a conventional Olsen testing machine. The loads are applied at the center of the beam, the respective tire deflections being measured with long-range dial extension-gages at each end of the beam. Usually, the load is applied in increments of 200 lb. from zero to 4000 lb. (2000 lb. per tire) and removed in like manner. These results give a load-deflection loop. After the car has been weighed, the normal load on the tire is determined. The average slope of the loop curve at this loading is taken as the tire rate. The difference in the loop at this loading is the static hysteresis of the tire. A moderate-size car usually has a tire rate in the range of 750 to 1100 lb. per in. Static hysteresis varies from 80 to 100 lb. A representative tire-rate curve is shown in Fig. 15.

The usual methods of determining the vertical and horizontal heights of the center of gravity were used, and it is not believed necessary to describe them here.

When the factors required by Professor Guest's analysis had been determined, it was possible to compute the location of the conjugate centers of oscillation P and Q and the natural periods about these centers. The results of such computations are given in Table 3 for six different cars of about the same weight and wheelbase. After completing such computations, the natural question arose as to whether or not they represented how the car body actually was moving. In an endeavor to check these computations experimentally, a set-up was made which is shown diagramatically in Fig. 16.

This arrangement consists of two heavy planks, one under the wheels on each side of the car, mounted on knife edges at one end and on an adjustable eccentric at the other. The car was rolled up on these planks until either the front or rear wheels were vertically above the center of the shaft carrying the eccentric, the knife edges were then placed vertically under the other pair of wheels. The eccentric shaft was rotated at increasing speeds until a speed was found which corresponds with the natural frequency of the car body. A light stiff board covered with regularly spaced dots, mounted on one side of the car, was photographed, using a time exposure, until the apparent center of rotation of the sprung mass was located; see Fig. 17. The data obtained from these photographs should give the location of the point P when the front wheels are oscillated and the point Q when the rear wheels are oscillated. It also should give the vertical height of the plane of the axis of oscillation (height, l, of the center of gravity) and the natural frequency about the two points P and Q.

Table 4 gives these data as obtained experimentally on six different cars of approximately the same weight and wheelbase compared with the calculated values given in Table 3. It will be noted that the agreement leaves much to be desired. In general, the greatest differences occur in the distances of the point P from the rear wheels and the frequency of oscillation about P. The discrepancies are greatest when the wheels over the stationary knife edge have leaf springs, Cars D, E and F, or the point P is far behind the knife edge. Under these two conditions friction in these springs may very markedly disturb the location of the point P or make it difficult to determine, because rotation about the points P and Q may not be independent. It is interesting to note that the vertical location of the line of centers through P and Q is in very good agreement with the calculated value. Further,

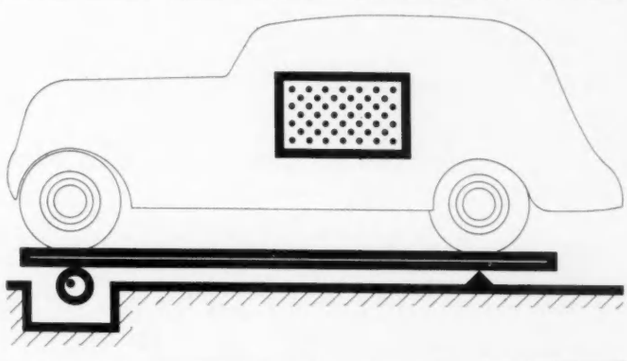
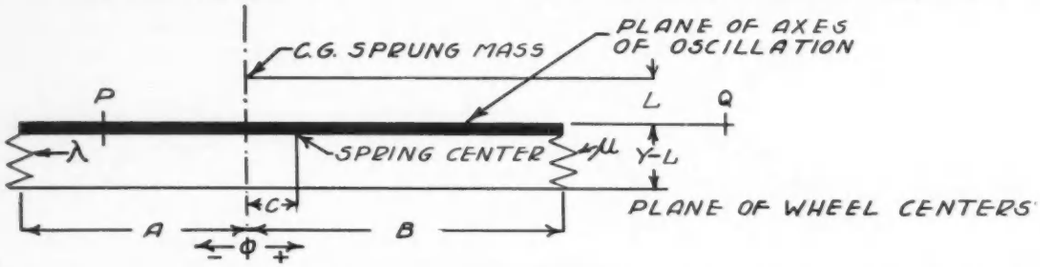


Fig. 16—Set-Up Used To Check the Computations Experimentally

Table 3—Computed Locations of the Centers of Oscillation and Frequencies About These Points



Car	Car Loading	Effective Spring-Rates Rear λ	Effective Spring-Rates Front μ	Plane of Axes of Oscillation Above Wheel Center-Line Y-L	Mass Constant (In².) $K^2 + L^2$	Elastic Constant (In².) $H^2 - LU$	Axes of Oscillation P from Rear Wheels	Axes of Oscillation Q from Front Wheels	Frequency About P	Frequency About Q
A	Road			11.8	2660	3140	−61	−35	80	92
Ride	1 Passenger	274	212	11.8	2520	3140	−101	−43	79	91
Fair	3 Passengers			12.1	2560	3140	43	415	84	76
B	Road			11.8	2690	3340	−157	−49	80	91
Ride	1 Passenger	277	216	11.8	2550	3340	−245	−54	78	90
Fair	3 Passengers			12.2	2590	3340	38	203	85	74
C	Road			12.1	2810	3140	−39	−28	69	78
Ride	1 Passenger	221	170	12.1	2670	3140	−63	−36	68	77
Good	3 Passengers			12.4	2730	3140	46	1114	71	66
D	Road			12.2	3430	3060	−8	−4	95	78
Ride	1 Passenger	238	435	12.2	3270	3060	2	0.9	94	77
Good	3 Passengers			12.0	3180	3060	6	4	94	71
E	Road			11.1	2890	2950	7	4	117	79
Ride	1 Passenger	250	437	11.1	2740	2950	11	7	116	77
Fair	3 Passengers			10.9	2710	2950	7	5	116	70
F	Road			11.2	3000	3170	29	65	81	77
Ride	1 Passenger	246	230	11.3	2850	3170	36	111	81	75
Good	3 Passengers			11.7	2840	3170	14	19	80	69

that in the case of Cars *A*, *B* and *C*, which had coil front springs, the agreement in frequency about the point *Q* at the front of the car is very good. These facts point to friction in the springs as the cause for the large discrepancies between the calculated and observed results. It might be found that a greater throw of the eccentric, that is, more than 1 in., would reduce the effect of friction. As the determination of natural frequency was observed independently of the location of center of rotation, it would be possible to compute one from the other and in this way obtain an additional check on the effect of friction.

Referring again to Table 3, a few interesting relations might be mentioned. The general opinion of quite a number of observers on the relative riding comfort of these cars rated them in the following order: *D*, *F*, *C*, *E*, *A* and *B*. Cars *D*, *F* and *C*, gave quite good rides; Cars *A* and *B*, rather poor. Some of the differences in the cars are, in all probability, due to differences in shock-absorber control. Cars *A*, *B* and *C*, used coil springs on the front and the usual stabilizer bar at the rear. Cars *D*, *E* and *F*, used leaf springs all around. It will be noted that the good-riding cars had the lower frequencies about *P* and *Q* and particularly about *Q*; that is, the point of rotation near the front axle which produces pitching in the rear seat. It is also interesting to note that, except for Car *E*, the ride is better the nearer the ratio of mass constant to elastic constant approaches unity. Table 5 gives these ratios for three different passenger loads. The high frequency about the point *P* in the case of Car *E* may disturb the ride.

It is also very probable that the relation of the seat location to the points *P* and *Q* is of considerable significance in the

Table 4—Comparison of Calculated and Observed Locations of *P* and *Q* and the Frequencies About These Points

Car	*A* Calculated	*A* Observed	*B* Calculated	*B* Observed	*C* Calculated	*C* Observed	*D* Calculated	*D* Observed	*E* Calculated	*E* Observed	*F* Calculated	*F* Observed
X-*L*	12	12	12	13	12	13	12	12	11	11	11	12
P From Rear Wheels	−62	−11	−157	+44	−39	+41	−9	+14	+8	+17	+29	+26
Q From Front Wheels	−36	−45	−50	−50	−29	−36	−4	+2	+5	+7	+66	+76
Frequency About *P*	80	90	80	89	69	82	95	100	117	121	81	96
Frequency About *Q*	92	91	91	91	78	78	78	86	79	82	78	92

Table 5—Comparison of Riding Quality with Mass; Elastic Constant Ratio

Car	Ratio	Ride Rating
A	0.81	5
B	0.78	6
C	0.86	3
D	1.04	1
E	0.92	4
F	0.90	2

ride. An examination of Table 3 also indicates that the closer the points *P* and *Q* are to the axle centers, the more satisfactory is the ride. It is also indicated by these data that the actual effective spring-rates of the front and rear springs do not by themselves control the comfort of the ride, the cars having the highest and the lowest spring-rates being among the three best rides.

The authors regret that it is not possible to draw any definite conclusions from the work described, but feel that the analysis presented by Professor Guest has made it possible to visualize clearly many of the problems connected with vehicle springing.

Appendix

The development of the formula used in calculating the moments of inertia obtained experimentally with a horizontal compound pendulum is given, to clarify further the method described in the foregoing text.

Consider any body, supported by a spring at a definitely located point and free to oscillate about another fixed axis as in Fig. 18. Taking moments about *O*, the center of oscillation, and assuming the spring to act in a vertical plane at all times, we find:

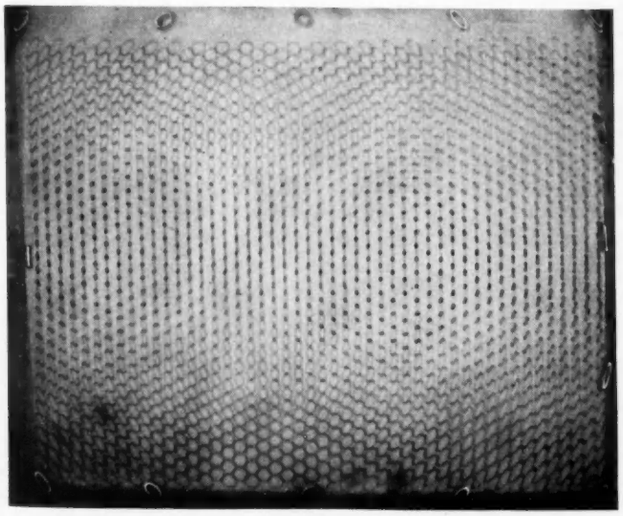


Fig. 17—Board, Covered with Regularly Spaced Dots, Mounted on One Side of the Car and Photographed To Locate the Apparent Center of Rotation of the Sprung Mass

Oscillations, 100 per min. Front-wheel lift, 0.63 in.

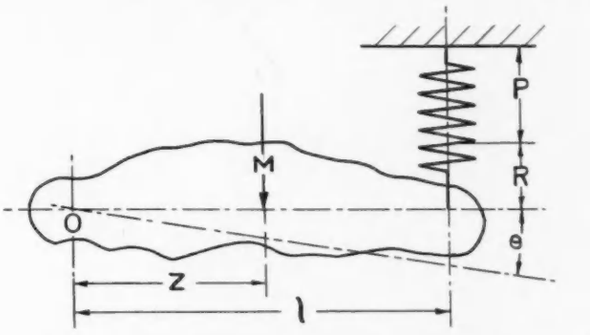


Fig. 18—Development of the Formula Used in Calculating the Moments of Inertia of a Horizontal Compound Pendulum

$$M_g Z - SLR = T \tag{1}$$

where

M = total mass
S = spring rate
T = total torque
R = extension of the spring under load

However, the total torque is $T = I\alpha$ which, differentiated, is:

$$T = I\frac{d^2\theta}{dt^2} \tag{2}$$

where θ is the angle of displacement.

Combining Equations (1) and (2), we have:

$$I\frac{d^2\theta}{dt^2} + SLR = M_g Z \tag{3}$$

Considering the action of this system, when disturbed, to be of a sinusoidal nature, the motion can be expressed by the equation:

$$\theta = \theta_o + A \sin 2\pi Nt$$

which, differentiated, is:

$$\frac{d\theta}{dt} = 2\pi NA \cos 2\pi Nt$$

$$\frac{d^2\theta}{dt^2} = -4\pi^2N^2A \sin 2\pi Nt \tag{4}$$

Combining Equations (3) and (4), we have:

$$\frac{d^2\theta}{dt^2} = \frac{M_g Z - SLR}{I} = -4\pi^2N^2A \sin 2\pi Nt$$

or

$$M_g Z - SLR = -4\pi^2N^2IA \sin 2\pi Nt \tag{5}$$

If a disturbing force is introduced into this balanced system equal to $SL^2\theta$, then Equation (5) becomes:

$$4\pi^2N^2IA \sin 2\pi Nt - SLR = -M_g Z + SL^2\theta \tag{6}$$

Since $\theta = A \sin 2\pi Nt$, then:

$$(-4\pi^2N^2I + SL^2)\, A \sin 2\pi Nt = M_g Z - SLR \tag{7}$$

Due to the condition of balance, $M_g Z = SLR$; therefore:

$$(-4\pi^2N^2I + SL^2)\, A \sin 2\pi Nt = 0 \tag{8}$$

$$4\pi^2N^2I = SL^2$$

$$I = \frac{SL^2}{4\pi^2N^2} \tag{9}$$

and

$$\frac{W}{g}K^2 = \frac{SL^2}{4\pi^2N^2}$$

or

$$K^2 = \frac{SL^2g}{4\pi^2N^2W}$$

Carbon Monoxide in Moving Vehicles

By F. M. Van Deventer
Cities Service Oil Co.

THIS paper reports upon the results of a highway investigation which justifies the conclusion that carbon monoxide in exhaust gases can and does seep into the passenger compartments of moving vehicles in sufficient quantities to impair the judgment of the driver in the control of his car; analyzes the cause factors resulting in the existence of carbon monoxide within cars; and recommends a form of inspection and maintenance which will effectively reduce, if not eliminate, this hazard to public safety.

Highway tests were made with the cooperation of the State police and motor-vehicle authorities in seven New England and Eastern States, several hundred cars being picked at random from State highways by the State police and examined, and between 5 and 6 per cent of the units tested were found to contain dangerous quantities of carbon monoxide in the passenger compartment. It was found that there were five forms of infiltration to account for the presence of carbon monoxide within cars.

Other phases of the subject are treated under the headings: Remedial procedure, asphyxial cases, comments on the foregoing, accident cases, general conclusions and recommendations.

THE fact that the exhaust gases from automotive engines contain sufficient carbon monoxide to produce asphyxial death does not constitute news. Between 500 and 700 such deaths occur in the United States annually, usually because an automobile engine is operated within a garage with closed doors, either intentionally or unintentionally. However, the fact that carbon monoxide in exhaust gases can and does seep into the passenger compartments of moving vehicles in sufficient quantities to impair the judgment of the driver in the control of his car has only recently been proved.

This paper reports upon the results of a highway investigation which justifies the conclusion just stated, analyzes the cause factors resulting in the existence of carbon monoxide within cars, and recommends a form of inspection and maintenance which will effectively reduce, if not eliminate, this hazard to public safety.

The manner in which carbon monoxide, when breathed, affects an average person is indicated in Fig. 1. The abscissa is hours of exposure and the principal ordinate is the per cent "CO" blood-saturation. Plotted upon the coordinates are a series of lines representing various concentrations of carbon monoxide in air breathed. The notes at the right side of the chart show the physiological effects associated with the CO blood-saturation. Thus, if from 25 to 35 per cent of the blood is carrying carbon monoxide, then the oxygen-carrying capacity of the blood is correspondingly reduced and the person may suffer headache, dizziness, or nausea as a result of oxygen deficiency. Whether or not these warning signs are observed, a person in this state of physical condition will not have the mental alertness which is required to reach instantaneous decisions in the proper control of a car. When approximately 40 per cent of the blood is carrying carbon monoxide the average person will collapse and, if not removed to fresh air, will die when approximately 50 per cent saturation is reached.

Coordinating the concentration of monoxide in air with physiological effects, it will be noted that one breathing air containing 16 parts of carbon monoxide per 10,000 parts of air would be in the zone of impaired judgment after an exposure of approximately ¾ hr., in the zone of collapse after 1 hr., and in the zone of probable death in 1½ hr. It must be understood that these effects represent an average adult in good physical condition. Anaemia and similar forms of illness accentuate the effect. Also, a child or adult of small stature is more rapidly affected.

It will be noted in Fig. 1 that very small concentrations breathed over a period of 2 or 3 hr. can also produce a state of impaired judgment. Thus, a concentration as low as 3

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parts of carbon monoxide in 10,000 parts of air may result in impaired judgment in about 1½ hr., although it is questionable whether a person breathing this concentration ever would collapse. It is considered, however, that 3 or more parts of carbon monoxide in 10,000 parts of air breathed constitutes a dangerous mixture, since it would impair the driving judgment of the operator of a car. The "3 parts per 10,000" may be visualized by pointing out that a glassful of carbon monoxide, when mixed with the air in the body of a coupe, will produce this mixture; and that this quantity of carbon monoxide is contained in a hatful of exhaust gases from a motor only slightly out of adjustment.

Prior to the winter of 1933, the Cities Service Co., in testing the exhaust gases of nearly 300,000 cars, had observed that the majority of the vehicles were being operated at a sufficiently low percentage of combustion efficiency so that they were producing excessive quantities of carbon monoxide in their exhausts. Aside from the fact that this constituted a waste of fuel, no further significance was attached to that fact. However, when the National Safety Council issued a public warning that there was an alarming increase in the number of drivers going to sleep behind the wheel, and when the Travelers Insurance Co. issued a bulletin indicating that there were nearly 500,000 "unexplainable" highway accidents

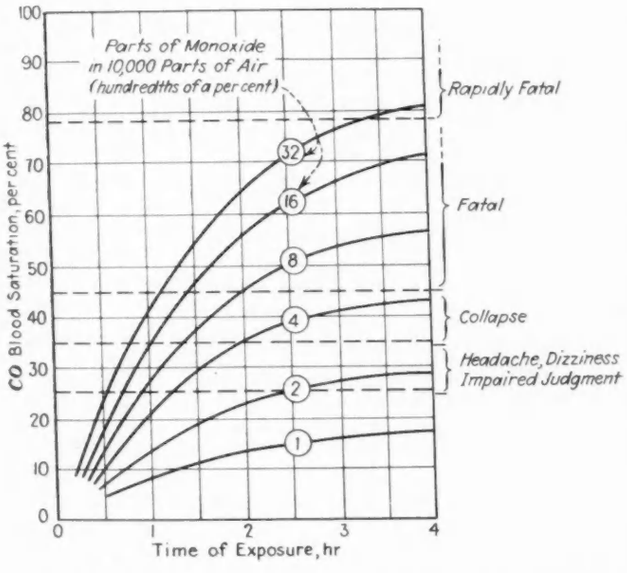


Fig. 1—Carbon-Monoxide Absorption by Human Blood

A person breathing air containing 8 parts of carbon monoxide per 10,000 parts of air probably would suffer headache and *impaired judgment*, and decreased mental alertness in about 1 hr.; collapse in about 1½ hr.; and, if not removed from the vitiated air, death in 2 hr. or more.

The chart represents a normal, healthy adult while essentially at rest. Anemia, or other ill health, and physical exercise accentuate the effects. Children or adults of small stature are more rapidly affected.

Individual susceptibility is an important factor. Gettler reports that the blood saturation accompanying death usually is between 45 and 65 per cent, but that isolated fatal cases have been encountered with as low as 28 per cent.

The physiological effects are based upon the examination of more than 2000 human cases by Gettler. The coordination between these effects and blood saturation has been endorsed by A. O. Gettler, chief toxicologist, Bellevue Hospital, New York City, C. D. Norris, chief medical examiner, New York City, and H. S. Martland, chief medical examiner, Essex Co., N. J.

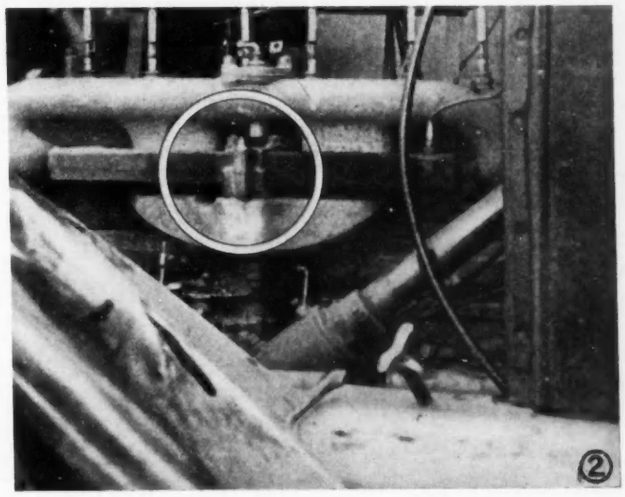


Fig. 2—Typical Exhaust-Manifold Gasket-Leak

each year, the observation concerning the prevalence of excessive carbon monoxide in automobile exhausts took on a new significance.

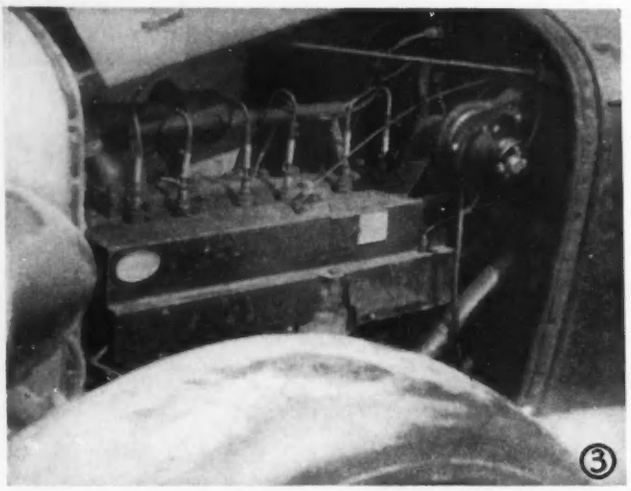
Highway Tests.—With the cooperation of the State police and motor-vehicle authorities in seven New England and Eastern States, several hundred cars were picked at random from State highways by the State police and examined.

The examination consisted first of driving the car on the highway and measuring the concentration of carbon monoxide in the passenger compartment by placing therein a Mine Safety Appliances Co. continuous carbon-monoxide indicator. This was followed by a determination of combustion efficiency and percentage of carbon monoxide in the exhaust using the Cities Service Power Prover. An examination was also made of the exhaust system to determine whether there were any leaking gaskets or flanges, split mufflers, or other defects which might result in the discharge of exhaust gases under the hood or floor boards of the car; and an examination of the floor boards and bulkhead separating the hood and body, to detect openings through which vitiated air might enter the body. The owner or operator of the vehicle was also asked whether he or other occupants of the car suffered headache, dizziness, nausea, drowsiness, or similar effects while riding in the car.

At the invitation of an association of fleet owners in New York City, a dozen or more representative units from each of several fleets were tested. Similarly, at the invitation of a highway official of one of the Central States, a group of more than 200 school buses was examined and tested. The following conclusions are based upon a tabulation covering all the work just referred to and is believed to represent the condition of the average run of cars on the highways today.

Results of the Highway Investigation.—Between 5 and 6 per cent of the units tested were found to contain dangerous quantities of carbon monoxide in the passenger compartment.

In 57 per cent, measurable traces or more of carbon monoxide were found. The significance of this finding is that, although these cars did not constitute a hazard at the time they were tested, the fact that a measurable trace was observed indicates that one or more of the several factors which contribute to the presence of carbon monoxide were already existent. The probabilities are that during continued use of the car these conditions will become worse unless attended to



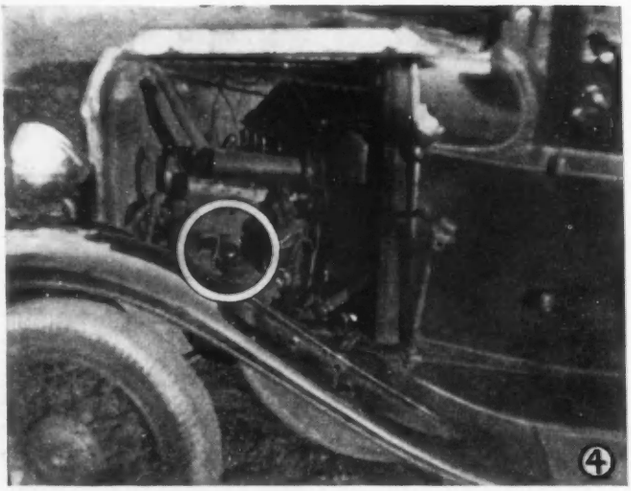


Fig. 3—An Exhaust-Manifold Heater Which Caused a Fatal Asphyxiation

Fig. 4—Crankcase Breather-Pipe Location Which Vitiates the Air Under the Hood

Fig. 5—A Fatality in Which the Driver Drove Off the Highway, Parked to Take a Nap, and Was Found Dead in His Cab

and that eventually the concentration of carbon monoxide will reach the danger level.

Detectable exhaust leaks at some point ahead of the normal tail-pipe discharge-point were observed in more than 40 per cent of the units tested. It should be explained that a majority of these leaks were not evidenced by the sound of escaping gases or by soot stains. Only leaks of major proportions are accompanied by these warnings. Many of them were observed by attaching a rubber hose to the sensitive carbon-monoxide detector and holding the free end of the hose adjacent to the several gaskets and flanges. In this way incipient leaks were found which accounted for small traces of carbon monoxide within the car.

Low combustion-efficiency which results in the production of excessive quantities of carbon monoxide in the exhaust gases was observed in 66 per cent of the units tested.

Forms of Infiltration

It was found that there were five forms of infiltration to account for the presence of carbon monoxide within cars:

(1) *Trailing Eddy Currents.*—It was found that following any but a perfectly streamlined car there is a vortical eddy current which has a tendency to entrain a portion of the exhaust gases from the tail pipe in the swirling current against the rear panel of the car. Since the slipstream at the sides of the car produces a suction within the body, some of the vitiated air trailing the car will be drawn in through any openings which exist in the rear panel. Station wagons and similar cars, in which a part of the rear panel is a curtain, constitute a typical example of this type of infiltration. In that case usually a high concentration is observed at the rear of the body, tapering off to a low and sometimes negligible trace at the driver's seat. In some cases, however, dangerous concentrations at the driver's seat result from this form of infiltration. The proper remedy is an approach toward streamlining and the elimination of openings such as that between the curtain and solid portion of the rear panel.

(2) *Pick-up Following Another Car.*—It was found that one car can follow another closely enough to pick up a dangerous portion of the exhaust of the leading car. In this case, the concentration of carbon monoxide within the second car depends upon the proportion of that gas in the exhaust and the amount of exhaust gas produced by the leading car. Thus, heavy vehicles such as trucks and buses produce a large quantity of exhaust gas, particularly at high speeds and on heavy pulls. If the second car be equipped with a type of ventilator or heater which draws "fresh" air from behind the radiator, the pick-up of exhaust gas from the leading car will be accentuated. In one instance during the tests, a car which was in itself in perfect condition became stalled in traffic behind a heavy vehicle, the resultant concentration of carbon monoxide within it was sufficient to affect the alertness of the occupants, and severe headaches resulted. The concentration was sufficiently high to have resulted in collapse and fatality in about 2 hr. While this was admitted an extreme case, there is no doubt that it happens frequently and may be the cause of many highway accidents.

(3) *Exhaust Leaks.*—Products of combustion containing carbon monoxide will issue from any loose gasket or other defective part of the exhaust system under the hood or body of the car. Fig. 2 shows a typical gasket-leak. The quantity of vitiated air under the hood leaking into the body of the car then will depend upon the relative tightness of the floor boards, bulkhead, and the like.

(4) *Exhaust-Manifold Heaters.* — Heater systems which operate on the principle of warming the air by making it flow over the exhaust manifold or muffler constitute one of the worst hazards observed during the investigation. When the manifold or muffler is thus inclosed, *all* the exhaust gas issuing from leaky gaskets or connections is *forced* into the body of the car. A number of asphyxiations have been directly traced to this cause. During the summer, a number of cars were tested in which exhaust heaters fortunately were turned off for the summer. Such cars were always tested with the heaters turned on and many were found which would have been death traps, inasmuch as a lethal mixture within the car would have obtained, the first time the heaters were used in the fall. Fig. 3 shows an exhaust-manifold heater which resulted in a fatal asphyxiation.

(5) *Crankcase Breather Pipes.*—Loose pistons and poorly fitted rings permit products of combustion to "blow-by" the pistons into the crankcase. With effective crankcase ventilation, these products will be discharged underneath the floor of the car. However, if the crankcase is vented through the oil-filler cap, Fig. 4, all the "blow-by" will be released under the hood, resulting in a vitiation of the air. While no dangerous conditions were found which could be attributed to breather pipes, many of the low concentrations of carbon monoxide observed were due solely to this cause. It can thus be a contributing factor which, when added to a minor gasket-leak, might result in a serious concentration.

Three Principal Factors.—It was found that all dangerous concentrations of carbon monoxide could be attributed to a combination of two or more of the following three factors:

(1) Low combustion-efficiency, which results in an excessive proportion of carbon monoxide in the exhaust gases

(2) A leak in the exhaust system

(3) Openings in floor boards or other parts of the body shell

No matter which of the five forms of infiltration are responsible for the existence of carbon monoxide within a car, it always follows that such concentration is in direct proportion to the percentage of carbon monoxide by volume in the exhaust gases of the engine producing them. The concentration within the car is also dependent upon the extent of the leak in the exhaust system and the extent of the opening in the car body.

Remedial Procedure.—It is obvious from the foregoing that inspection and preventive maintenance constitute the proper means of controlling and minimizing the dangers of carbon monoxide. By maintaining a high combustion-efficiency, the carbon in the fuel will be almost completely burned to carbon *dioxide* within the cylinders and there will then be a minimum of carbon *monoxide* in the exhaust gases. At the same time, economy and performance will be at their best.

When new cars are delivered from the factory they are, or should be, adjusted to operate at a high combustion-efficiency. However, a periodical checking of the various precise adjustments of the ignition and carburetion systems is required to maintain proper combustion-efficiency during the life of the engine. If all engines were thus continuously maintained and if the exhaust systems and car bodies were kept tight, the carbon-monoxide dangers would be minimized, if not completely eliminated.

The following examples of several actual cases will serve to illustrate how low combustion-efficiency and leaks resulted in asphyxiations or accidents.

Fig. 6—Broken Seam in the Muffler of the Truck Shown in Fig. 5

Fig. 7—A Defective Exhaust-Pipe Flange

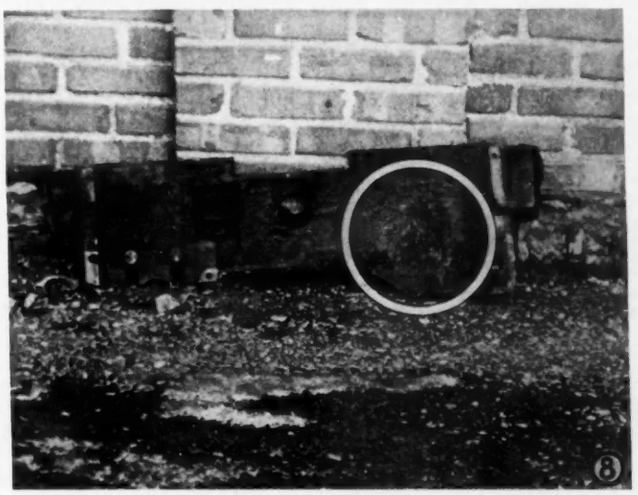


Fig. 8—Loose Floor-Boards Charred by Gases from the Defective Muffler Shown in Fig. 6

Asphyxial Cases.—The operator of the truck shown in Fig. 5 died behind the wheel. While dead men tell no tales, a half-empty box of headache tablets and footprints in the snow around the car bore mute evidence that the driver had suffered headaches before the tragedy. Apparently, his truck struck a guard rail and the driver, realizing that he was "groggy," had pulled off the pavement for the purpose of taking a nap. He was found several hours later seated at the wheel with the engine still running. When the carbon-monoxide detector was placed within the cab and the engine started, it showed a concentration of carbon monoxide which would cause the death of any normal person after only a brief exposure. It was not feasible to test the car on the highway, but there is little doubt that the concentration then would have been sufficient to result in drowsiness, if not complete collapse. The Power Prover showed that less than 50 per cent of the fuel was being burned within the engine and that the exhaust gases contained more than 15 per cent of carbon monoxide by volume. The exhaust system contained several leaks, the worst of which was a rip in the seam of the muffler as shown in Fig. 6. A less-important leak occurred at the flange connecting the exhaust manifold and exhaust pipe. The crystallization and crack at the flange, as indicated in Fig. 7, is typical of the condition in which similar flanges in many cars were found, some of which resulted in dangerous carbon-monoxide levels within the cars.

The floor boards were loosely fitted and without the protection of a mat. Exhaust gases from the split muffler had impinged upon the underneath side with sufficient force to char the wood, as shown in Fig. 8.

In Fig. 9 is shown a car in which two occupants were asphyxiated while the car was parked. Fig. 10 shows the detector placed within the car so as to be observed through the window, and the State officials who witnessed the test. A lethal atmosphere within the car was indicated within 5 min. after the engine was started. The Power Prover indicated a combustion efficiency of less than 50 per cent and a consequent high percentage of carbon monoxide in the exhaust. The car was equipped with an exhaust-manifold heater, shown in Fig. 3, and, when this was removed, the exhaust gasket-leak indicated in Fig. 2 was discovered. It is interesting to note that the leakage from this gasket was not

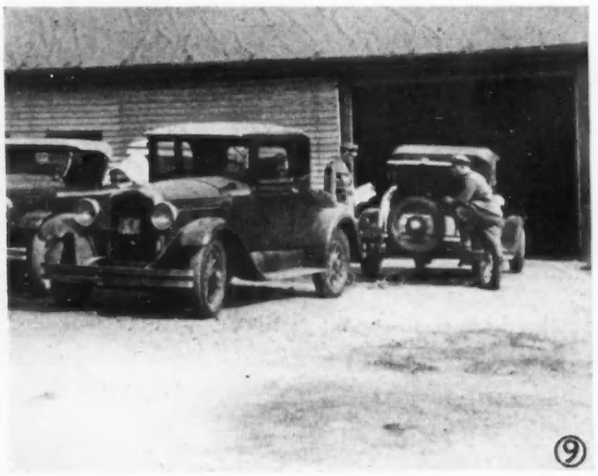


Fig. 9—A Car in Which Two Occupants Were Asphyxiated

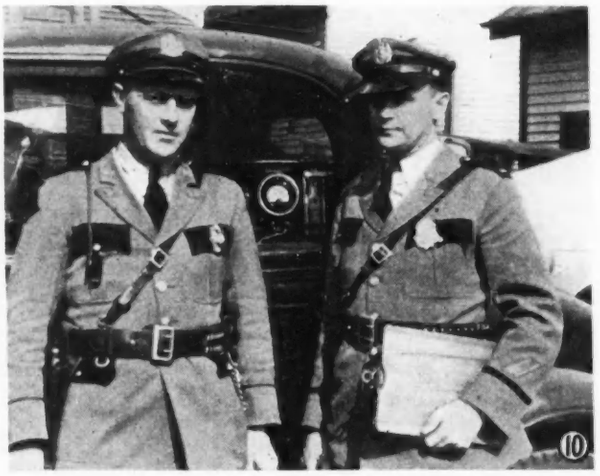


Fig. 10—The Carbon Monoxide Detector Installed in a Car; Test Witnessed by State Police

Fig. 11—A Fatality in Which a Car Was Driven into the Ditch and the Owner Was Asphyxiated While Trying To Regain the Highway

Fig. 12—Gas Mask Worn While Driving Cars in Which Deaths Had Occurred

sufficient to cause an audible sound and also that the air within the car, even though of a lethal concentration, contained no warning odors. This was a splendid example of the insidiousness of carbon monoxide. The windshield of this car bore a sticker indicating that it had passed the State compulsory examination of the "safety devices" only a few weeks prior to this tragedy.

In Fig. 11 is shown a car, the driver of which drove off a straight road into the ditch where the car was found in the position indicated by the police officer in the picture. When found several hours later, the owner was dead, slumped over the wheel. To determine the concentration of carbon monoxide within the car, which apparently was responsible for the driver leaving the road, the car was operated over the route which the owner had previously traversed. The detectors were placed on the front seat and the investigator wore an All-Service gas-mask, as indicated in Fig. 12, to protect him from the carbon monoxide present. When driven on the highway, concentrations as high as 10 parts of carbon monoxide in 10,000 parts of air were indicated, which would seemingly account for the fact that the driver lost control of his car after only a moderate drive. With the engine idling and the car stationary, as it was when found, the concentration reached 30 parts per 10,000, which would account for the operator's death.

Exhaust Leak Caused Death

The Power Prover indicated a combustion efficiency below 50 per cent and an exhaust mixture excessively rich in carbon monoxide. The exhaust leak occurred at the flange between the manifold and exhaust pipe, although the owner had performed a "hay-wire" repair, as indicated in Fig. 13. The floor boards were loosely fitted, were not covered by a mat, and contained several openings of major proportions. The moral of this case is: "A gasket is cheaper than a casket."

Comments.—It will be observed that, in each of the three cases which have just been described, all three of the principal factors contributed to the conditions which caused the several asphyxiations. The exhaust gases contained abnormally high quantities of carbon monoxide due to maladjustments of the ignition or carburetion systems. If the engines had been properly tuned to a combustion efficiency of 85 per cent, the carbon monoxide within the passenger compartments would have been reduced to at least one-fifth of the observed concentrations. In each case there was a leak in the exhaust system and an opening in the car body through which the vitiated air could enter.

Accident Cases.—It is not often feasible to examine cars in which carbon monoxide caused an accident, because usually such cars are damaged beyond the possibility of driving them on the highway. Two examples, however, will serve to illustrate accidents which are believed to be typical of thousands in which carbon monoxide was a contributing, if not direct, cause-factor.

The driver of a truck, after having been on the highway between 2 and 3 hr., drove head-on at moderate speed into the rear of another parked truck. While the vehicle was damaged beyond the possibility of determining the source of carbon monoxide, an examination of the dead-driver's blood indicated a carbon-monoxide saturation which would seem to account for his poor driving-judgment and lack of alertness.

The owner of a private vehicle, after driving about 2 hr., drove into a street-car "Safety Loading-Zone." One person

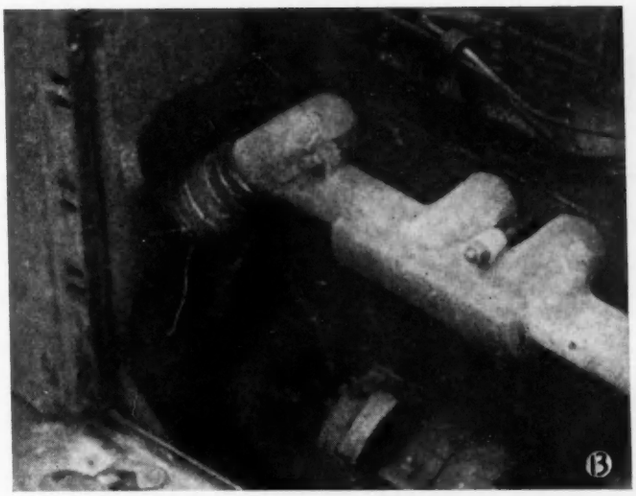


Fig. 13—"Haywire" Repair of a Defective Exhaust Flange in the Car Shown in Fig. 11

was struck down and killed, and two others would have been killed, had they not seen the impending danger and jumped. The operator drove his car well into the intersection before bringing it to a stop. He was at first held for "driving while intoxicated" as well as for manslaughter and reckless driving, because he was observed to stagger when alighting from the car. Medical examination, however, disclosed absence of alcohol but presence of carbon monoxide in the driver's blood.

Conclusions and Recommendations

General Conclusions.—In the light of the foregoing examples of what can happen when dangerous quantities of carbon monoxide exist in automobiles and the percentage of those cars tested which contained dangerous quantities of this gas, it would seem that there may be half a million or more cars in operation on the American highways which are definite hazards to public safety, and that effective steps should be taken to eliminate or at least minimize this danger.

Recommendations.—It is recommended that a system of periodical inspection be provided for whereby all cars will be tested as to their combustion efficiency and examined as to the maintenance of their exhaust systems and bodies. This could most effectively be accomplished by adding these examinations to the compulsory inspection of other safety devices as now provided for in some States.

It is further recommended that the public be familiarized with the symptoms and dangers of carbon monoxide so that when persons experience headaches in a car they will have the car examined rather than buy headache tablets and continue the drive.

The army marching-custom, whereby five minutes out of each hour is devoted to a rest period, becomes a splendid driving rule. If persons on long drives would spend five minutes out of every hour exercising in the fresh air, their journey would be not only safer but more pleasant because such a rest period would prevent muscular and mental fatigue and tend to replace with oxygen any carbon monoxide which the blood had accumulated during the preceding hour.

High-Speed C-I Engine Performance; Three Types of Combustion Chamber

By Ernest G. Whitney

Assistant Mechanical Engineer, National Advisory Committee for Aeronautics

MIXING of the fuel and air presents the major problem in maintaining high efficiency at low excess-air percentages in the high-speed compression-ignition engine. The outputs obtained from a single-cylinder 5 by 7-in. test-engine with three types of combustion chamber are compared and their respective characteristics discussed.

Airflow is depended upon for mixing the fuel and air in the pre-chamber engine. Performance is shown to be influenced by clearance distribution, connecting-passage size, pre-chamber shape, location of the fuel spray, and boosting.

The integral, or quiescent combustion chamber depends upon multiple sprays properly proportioned and directed to reach the available air. Results of an extensive series of injection-nozzle variations are shown in tabular form.

The effects of scavenging and boosting and of high coolant-temperatures are discussed. Optimum performance showed either type to be inadequate for use as an aircraft powerplant.

The displacer-piston combustion-chamber combines the advantages of directed multiple fuel-sprays and forced airflow and gave performance definitely superior to either of the other two types, allowing smokeless exhaust to be obtained with only 15 per cent excess air.

Effects of passage width, displacer height, displacer shape, optimum fuel-spray, high-temperature coolant, and boosting are shown. Friction is discussed briefly. Specific outputs are shown to be comparable with those obtainable from boosted gasoline engines. Further improvement in performance is predicted.

THE demand of automotive apparatus, and aircraft in particular, for a powerplant of high output, small bulk, low weight, and efficient operation has been most ably met by the modern internal-combustion engine. Dependable and durable air-cooled gasoline-engines are available which develop 1 hp. for less than 1 1/3-lb. of weight and which may be operated continuously at a large fraction of full power at a specific fuel consumption of 0.48 lb. per b.hp-hr. These outputs and economies are direct results of contemporary developments of materials, engine design, and fuels. The high-octane gasolines recently made widely available allow the engine manufacturer to boost large cylinders with compression ratios of 6.5:1 and 7:1, thereby realizing in large engines the better efficiencies accompanying the high expansion-ratios. Further increases in output and economy would follow the availability of still better fuels, as has been proved by tests with gasolines of desirably high octane number. The possibility, however, of such better fuels being made widely available is rather remote inasmuch as the supply of suitable natural products is limited and economic considerations forestall their synthetic preparation. It seems unlikely, therefore, that future large improvements in engine performance will result from this source.

The compression-ignition or so-called "Diesel" cycle, on the other hand, offers the possibility of even better economies without the fuel limitation. On the contrary, the fuel required is marketed widely and at a much lower price than the subject gasolines. Very high expansion-ratios make more work available at the piston and decrease the problem of waste-heat dissipation. As a consequence, for a given power-output, the compression-ignition engine requires less fuel of lower cost than does the gasoline powerplant. To the aircraft operator, the elimination of the ignition system and material reduction in fire hazard make the compression-ignition engine additionally attractive.

The problems that have prevented the compression-ignition engine from supplanting the gasoline engine have been:

(1) Utilization of available air charge. Lower specific outputs result from the necessity for excess air to maintain clear exhaust

(2) High mechanical losses

(3) High cylinder-pressures, necessitating heavy engine-parts

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 19, 1935.]

The first two points will be discussed in the light of research conducted at the Langley Field laboratory of the National Advisory Committee for Aeronautics. Inasmuch as the Committee's interest in the compression-ignition engine concerns its usefulness as an aircraft powerplant, the problem of obtaining maximum specific output assumes paramount importance. Accordingly, most of the discussion will be devoted to the performance obtained with different systems of air-fuel mixing obtained by the use of three types of combustion chamber. Much of the material presented has already been published and may be found by reference to the N.A.C.A. reports and notes listed in the appended bibliography. More recent information will become shortly available in the same form.

General Problem

The chief factor hindering the mixing of the fuel and air in the compression-ignition engine is the shortage of time. In the carburetor engine, two full strokes are available during which the air and gasoline vapor may become perfectly distributed. In the compression-ignition engine, these two strokes are used solely for the induction and compression of the air charge. During the period used by the spark-ignition engine for the burning of the distributed mixture, the Diesel engine must inject its fuel charge, distribute the charge evenly throughout the compressed air, and complete the combustion. The time at which the fuel is introduced and the rate of burning must be controlled to prevent the too-rapid rise of gas pressures and the attainment of excessive maximum pressures. At a rotative speed of 2000 r.p.m., about 0.0015 sec. is available for the introduction of the fuel charge. At 0.0025 sec. later, the fuel should have sought out and combined with all the oxygen present, leaving the remainder of the stroke for expansion. The development of means to accomplish this end would give the compression-ignition engine a large advantage over the spark-ignition engine in power output.

Investigations by the National Advisory Committee for Aeronautics have resulted in gradual improvement in compression-ignition-engine performance by using increasing proportions of the available air charge. Clear exhaust can now be obtained at an engine speed of 1500 r.p.m. with 15 per cent excess air. Maximum indicated mean effective pressures (unboosted) of 165 lb. per sq. in. have been obtained. No reason has presented itself for the belief that continued study should fail further to reduce the excess-air requirement and correspondingly increase the output.

Tests have been conducted on 5 by 7-in. single-cylinder test-engines (Fig. 1) with three types of combustion chambers: the pre-chamber, the quiescent, and the displacer-piston types. The chambers will be discussed in the order named.

Pre-Chamber Type of Combustion Chamber

The pre-chamber is a form of divided combustion chamber in which part of the clearance volume is formed in an auxiliary space separated from the cylinder proper by a small connecting passage. Fuel enters the cylinder through the pre-chamber, the high-velocity forced airflow through the small connecting passage being relied upon to effect mixing. This type of combustion chamber is used successfully in certain American and foreign commercial Diesel engines and has several desirable characteristics, being simple of construction and capable of fairly good outputs with a plain single fuel-injection spray. It is, however, hard to start, induces high friction losses, and is prone to combustion knock unless inefficiently small connecting passages are used. Specific output and economy are inferior to those of the displacer-piston type, as will be shown later, and it is therefore not suited to aircraft-engine use.

Fig. 2 shows the experimental cylinder head assembled with a disc-shaped pre-chamber and tangential connecting passage. The chamber cap and passage insert are removable or ex-

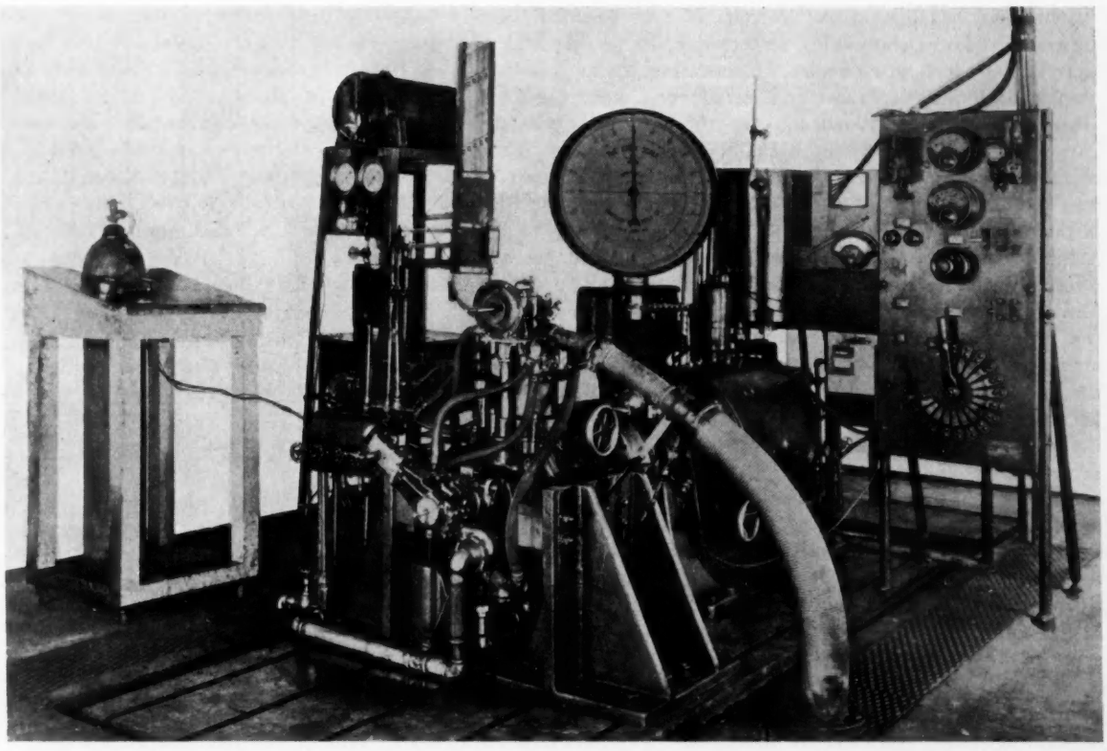


Fig. 1—Single-Cylinder Research-Engine and Testing Equipment

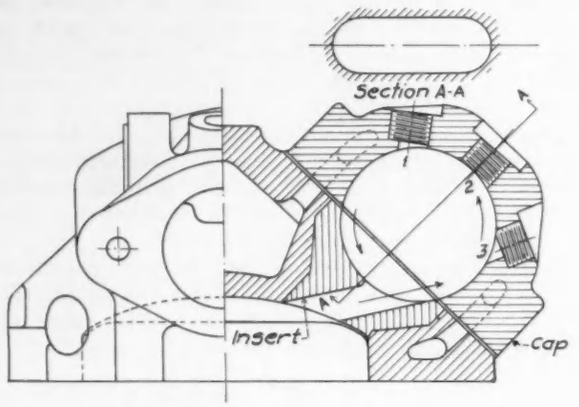


Fig. 2—Pre-Chamber Cylinder Head

changeable to permit investigations of sizes, shapes, and passage direction. Three optional locations for the injection valve are shown at 1, 2, and 3.

Rather comprehensive tests have been conducted with the pre-chamber engine, covering, in all, a period of over eight years. From a study of accumulated data and comparison with other test results, the limitations of the pre-chamber have become apparent. The most important investigations will be briefly reviewed. They include studies of clearance distribution, connecting-passage diameter, chamber shape, compression ratio, and boosting.

Clearance Distribution.—For a divided combustion chamber, there must obviously exist an optimum relation between the compression volume retained in the cylinder and that forced into the pre-chamber. For a chosen total clearance volume (13.5:1 compression ratio), the optimum clearance distribution was determined by tests of four spherical combustion chambers containing, respectively, 20, 35, 50, and 70 per cent of the total clearance. Connecting passages of constant length-diameter ratio and flared ends were used, the diameters being calculated to give the same air velocities through the passages for each chamber size. Passage diameters were 23/64, 15/32, 9/16, and 43/64 in. The passage axis passed through the centers of the spheres and coincided with the axis of the fuel spray.

Fig. 3 shows the manner in which engine performance was affected by variation in clearance distribution. The output and economy improved as the percentage clearance volume in the pre-chamber increased, presumably because the available air was better disposed for mixing with the fuel spray. With the volume in the pre-chamber below 35 per cent, performance was particularly inferior. The 35-per cent chamber gave the least knock and best combustion control at some sacrifice in power.

Measurements of heat loss to the cooling water showed an increase in total loss from 20.5 to 28.5 per cent of the total heat in the fuel as the chamber volume increased. This increase was caused by higher combustion temperatures in the chamber and also by an increase of approximately 10 per cent in the total combustion-chamber surface-area. As would be expected, the amount of heat loss from the chamber cap increases with chamber volume and surface; whereas the loss from the head decreases. An increased percentage of loss to the cylinder jacket-water for the 20-per cent pre-chamber indicated late burning and served as a further proof of inferior fuel and air mixing obtained with that chamber.

For the passage size selected, the indicated mean effective pressure was maximum at an engine speed of 1200 r.p.m. for all clearance distributions tested. Friction power increased linearly with speed and very slightly with increasing pre-chamber volume, resulting in a maximum brake torque developed at somewhat lower speeds.

Connecting-Passage Size.—Variation of the connecting-passage diameter using a spherical pre-chamber containing 50 per cent of the clearance volume gave the motoring characteristics presented in Fig. 4. In general, the friction losses decreased with increase in passage size. For diameters less than 29/64 in., the throttling was excessive as shown by the rapidly increasing friction and the pressure differential between the cylinder and combustion chamber. Starting, idling, and combustion sound, however, were superior with the smaller passage sizes. As the diameter was increased, the knocking intensity increased up to a passage diameter of 17/32 in., beyond which the knock became less intense.

The effect of passage size on power performance and specific fuel consumption at an engine speed of 1500 r.p.m. is shown by Fig. 5. At this speed, the optimum passage diameter is found to be between 29/64 and 17/32 in. Data show the ignition lag to be decreased and the combustion process

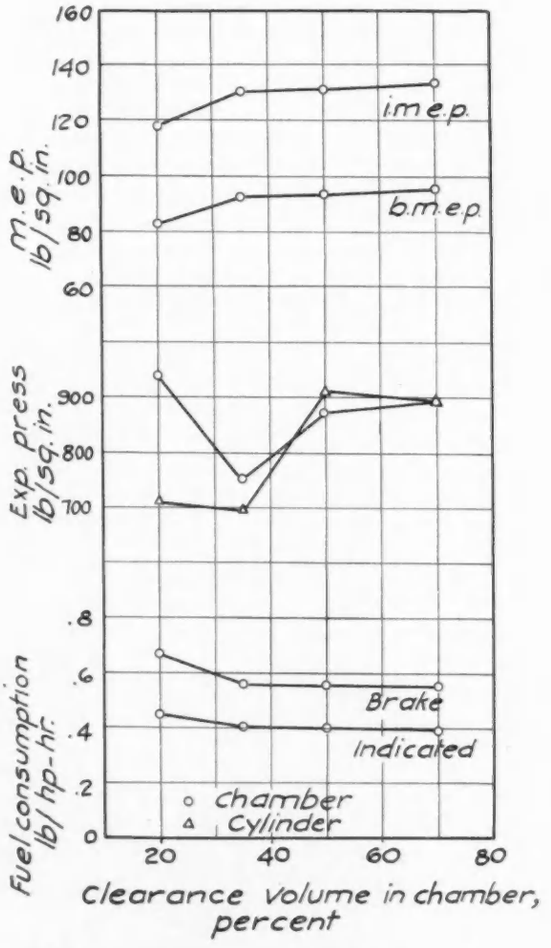


Fig. 3—Effect of Clearance Distribution on Pre-Chamber Engine Performance at 0.0003 Lb. of Fuel per Cycle; 12 Per Cent Excess Air

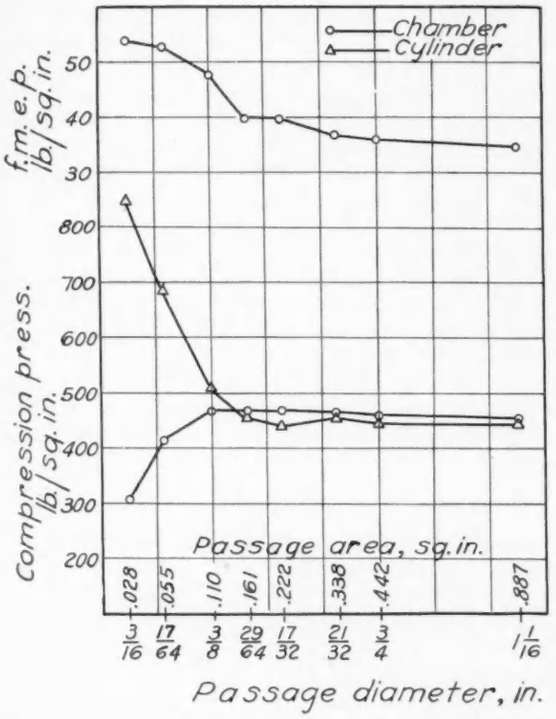


Fig. 4—Effect of Passage Diameter on Motoring Characteristics of Pre-Chamber Engine; 1500 R.P.M.

speeded up by the use of small passages. Engine speed, however, determines the air velocities through any given passage area and the resultant friction losses. Hence, the optimum passage size for any speed will be determined as the best compromise between these opposing influences.

Chamber Shape.—Substituting a disc-shaped combustion-chamber for the spherical volume and introducing the connecting passage tangentially to the lower edge of the disc gave the assembly shown in Fig. 2. This combination with 50 per cent of the clearance in the disc conserved the forced airflow into residual flow and gave performance superior to that obtained with the spherical chamber of similar volume. The general operating characteristics—starting, idling, cyclic regularity, and combustion shock—were little affected. Later tests showed that starting could be made easier by injection directly into the cylinder.

Rounding the cylinder end of the connecting passage was found to increase the performance, presumably by allowing the issuing jet to spread throughout more of the cylinder clearance. Rounding the chamber end of the passage evidently decreased the effective air-swirl and decreased the performance somewhat. Variation of the passage direction into the cylinder showed the performance to be optimum when the passage entered radially.

The location of the injection valve in Position 1 (Fig. 2) so as to inject almost at right angles to the airflow gave best performance; injection from Position 3 directly into the air jet gave the poorest. Combustion of the spray from a single 0.050-in. orifice was equal or superior to that of a conical or multi-orifice fan-shaped spray. The optimum injection period was found to be 21 deg.

A solid-metal spool inserted into the disc-shaped chamber changed it to a toroid to intensify the airflow still further by removing the air in the low-speed center-section of the disc. Performance with this chamber was inferior, however, probably because of the interference with the airflow and fuel spray by friction and impact.

Fig. 6 shows the mean effective pressures, explosion pressures, and specific fuel consumptions obtained with the 50-per cent disc pre-chamber, having a single tangential connecting passage flared at the cylinder end, and a single 0.050 spray in the optimum location for various quantities of fuel injected. These curves may be taken to approximate the optimum performance to be expected from the pre-chamber type of cylinder head. A slight improvement could be obtained by enlarging the pre-chamber to include a greater percentage of the clearance volume.

Compression Ratio.—Using the disc pre-chamber, engine performance was compared at compression ratios of 13.5, 14.5, 15.5, 16.5, and 17.5. For convenience of investigation, the ratios were changed by decreasing the cylinder clearance so that the corresponding percentages of total clearance volume contained in the pre-chamber were 50, 54, 58, 62, and 66, respectively. Results showed the full-load indicated mean effective pressure to increase slightly with compression ratio, approaching a maximum at 17.5:1. The increase of four ratios netted an increase of but 4 lb. per sq. in. in mean effective pressure. Inasmuch as the engine friction also increased with compression ratio, the full-load brake-performance remained substantially unchanged. The only improvements to be realized by the use of very high compression ratios were found to be easier starting and reduction in combustion knock. The most desirable ratio was selected as 15.5:1.

Boosting.—To determine the increase in output to be ob-

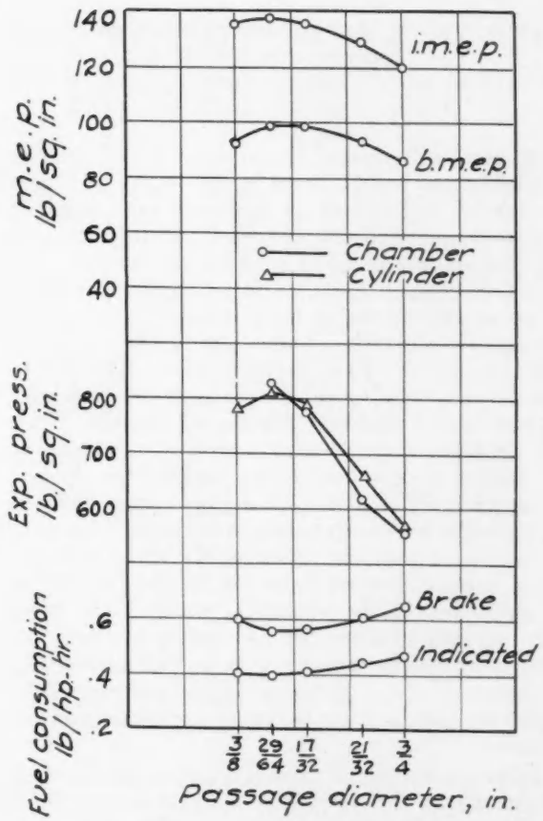


Fig. 5—Effect of Passage Diameter on Pre-Chamber Engine Performance at 0.000325 Lb. of Fuel per Cycle; No Excess Air; 1500 R.P.M.

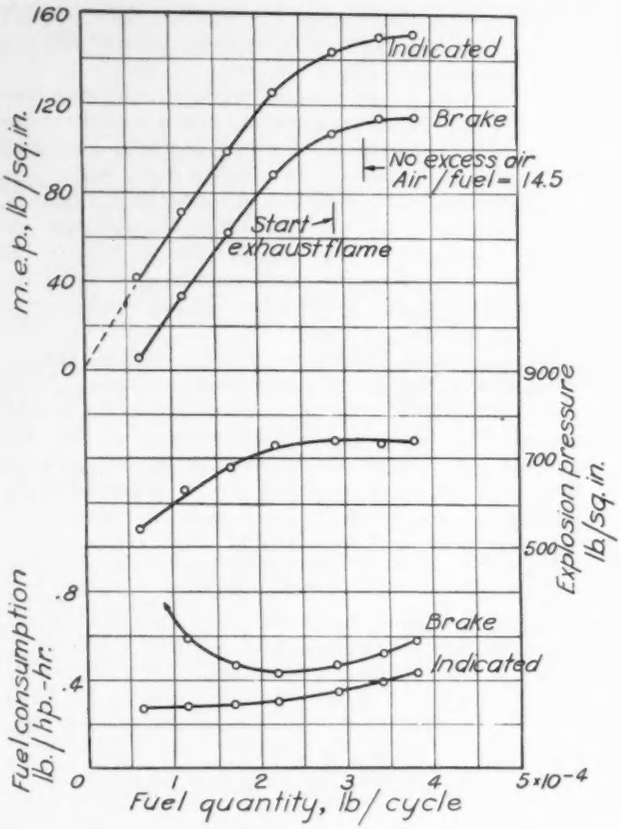


Fig. 6—Effect of Fuel Quantity on Pre-Chamber Engine Performance at 1500 R.P.M.

The 50-per cent disc chamber single tangential connecting passage, and the 0.050-in. injection-nozzle.

tained from an increased air charge, the best pre-chamber combination was boosted to 2½, 5, 7½, and 10 in. of mercury. Fig. 7 shows the indicated and net brake mean effective pressures, explosion pressures, and fuel consumptions obtained under the four conditions of boost for various fuel quantities. The indicated performance is seen to improve throughout the fuel range with increasing boost without imposing excessive explosion pressures. Net brake performance (corrected for power required to supply the air) suffers, however, at fuel quantities below 4.25 by 10-4 lb. per cycle, in which range the increased friction overcomes the thermal advantage. A maximum net brake mean effective pressure of 143 lb. per sq. in. was obtained at 10 in. of mercury boost. With the boosting air independently supplied, the engine friction was constant over the range of boost pressures investigated, indicating that the decrease in pumping losses usually found with boosted engines is more than offset by the increased throttling loss through the pre-chamber connecting passage. The additional power required to drive the air charger increases the net friction mean effective pressure with increasing boost pressure.

Conclusions.—From the foregoing brief summary of the pre-chamber investigations, the following general conclusions may be drawn:

(1) The pre-chamber is capable of giving rapid mixing and combustion with fuel injected in a single compact injection spray from a large round-hole-orifice nozzle.

(2) The pre-chamber should be disc-shaped and contain as much of the clearance volume as possible. The connecting passage should be as small as is consistent with the maximum designed engine speed without allowing the attainment of acoustical air velocities. The passage should enter the pre-chamber tangentially to give a strong rotational airflow and should flare into the cylinder radially. The injected fuel spray should be directed across the pre-chamber at right angles to the entering airstream; an injection period of 21 deg. should be used. The compression ratio should be about 15.5:1.

(3) Boosting the optimum pre-chamber with no other change will increase the net output, an increase of about 2½ lb. per sq. in. brake mean effective pressure resulting from each 1 in. of mercury boost below 10 in. Specific fuel consumption at part loads will be inferior because of the added power absorbed by the blower.

(4) Hard starting, high friction losses, presence of combustion knock, and inability to scavenge the combustion chamber eliminate the pre-chamber engine as a desirable aircraft powerplant.

Quiescent Type of Combustion Chamber

A second type of combustion chamber investigated is the vertical-disc type shown in Fig. 8. As there is no restricting orifice between the chamber and cylinder proper and as no evidence has ever presented itself to indicate that combustion was affected by an air movement, this cylinder head has been referred to as a "quiescent" combustion chamber. The absence of air movement leaves the burden of distribution and mixing of the fuel and air to the injection system alone. For optimum performance, the injected spray must be so disposed and so

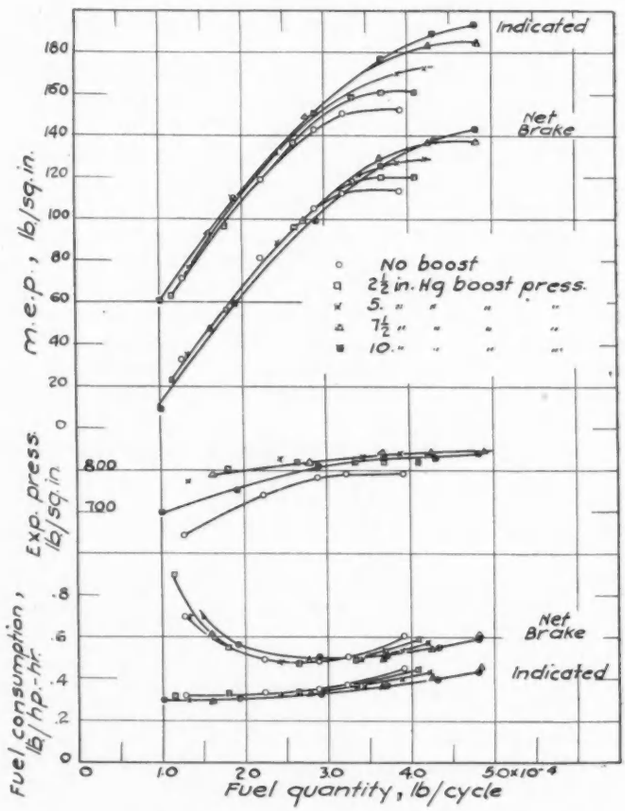


Fig. 7—Effect of Boost Pressure on Pre-Chamber Engine Performance at 1500 R.P.M.

The 50-per cent disc chamber, the single tangential connecting passage, and the 0.050-in. injection-nozzle.

proportioned as to seek out all the available air in the proper proportions to give homogeneous mixing. Obviously, more sprays than the single jet of the pre-chamber will be required.

Nozzle Investigation.—Early tests of available types showed the multi-orifice nozzle represented by Fig. 9 to offer the best possibilities of accomplishing the desired result. Accordingly, the multi-orifice nozzles with an automatic injection valve were made the basis for extensive investigations of orifice number, size, and angular spacings.

To describe in detail all the orifice combinations tested and results obtained would be a task almost as endless as the experimental investigation itself. In general, the problem was attacked by two methods:

(1) By proportioning and directing the fuel jets to serve zones of air in a calculated correct ratio

(2) By systematically varying orifice sizes, number, and angles to cover the most likely combinations.

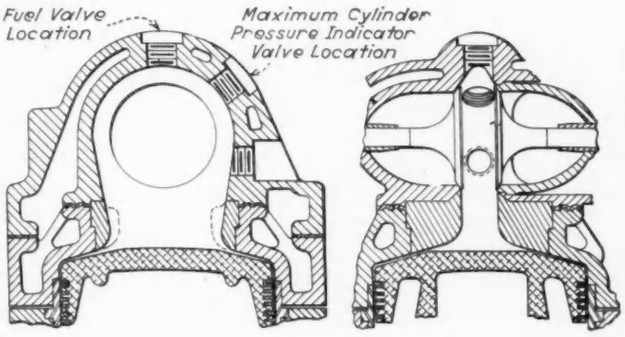


Fig. 8—Quiescent Combustion Chamber

Neither method in itself is complete. The first presupposed orifices of geometrical similarity and an injection system so designed to make the same pressure effective in causing the discharge through each orifice. The second relied upon performance to indicate trends and required careful consideration of data inasmuch as the same performance values might be attained by different combinations, one of which representing the optimum of a particular means of improvement whereas the other fell on an intermediate point along a line leading to possibilities of even better performance. A judicious combination of the two methods assisted by a study of spray photographs offers the quickest approach to a determination of the optimum arrangement.

Table 1 summarizes the general orifice groups investigated with representative variations tested for each group and is self explanatory. For each orifice arrangement the total orifice-area, individual-orifice diameters, percentage of total orifice-area represented by each orifice, indicated mean effective pressures, indicated fuel consumptions, maximum cylinder pressure, and percentage of full-load fuel burned with clear exhaust are given. For all except the *K*-4, the *M*-1, and the *M*-2 nozzles, performance data were obtained at 1500 r.p.m. engine speed and 12.6:1 compression ratio with injection advanced to give a "permissible knock." The latter three nozzles were tested with the combustion chamber shortened to raise the compression ratio to 15.3:1. Higher cylinder pressures and better performance could then be obtained.

Comparison of the results of Table 1 with Fig. 6 shows the

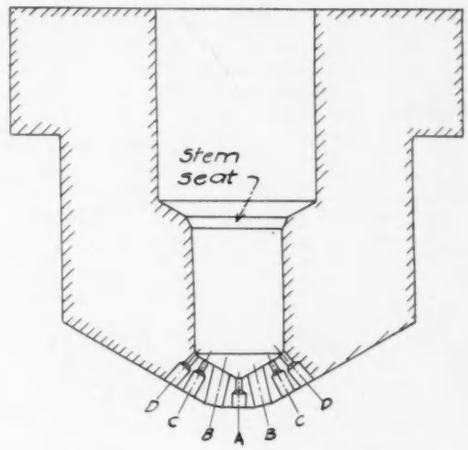


Fig. 9—Multi-Orifice Injection-Nozzle

full-load output of the pre-chamber to be 9 per cent greater than that of the quiescent combustion chamber, the advantage being gained, however, at the expense of more objectionable knock and harder starting. Friction mean effective pressure of the quiescent combustion chamber was 4 lb. per sq. in. lower than that of the pre-chamber.

Scavenging and Boosting.—By virtue of the location of the inlet and exhaust valves in the combustion chamber, it was found possible further to improve the output of the quiescent

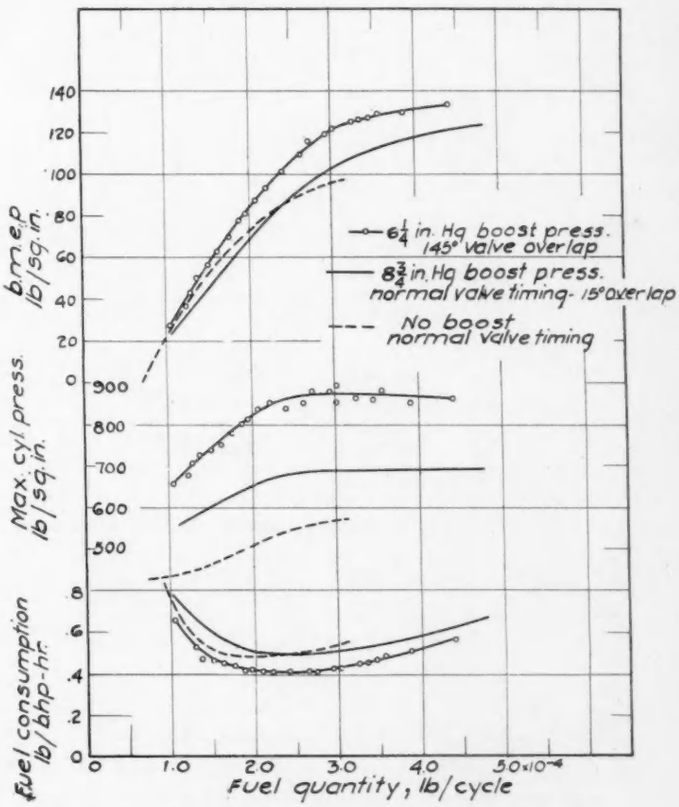
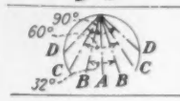
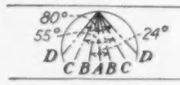
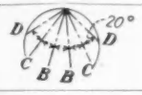
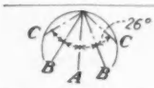
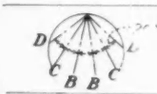
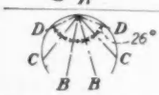

bm.e.p lb/sq.in.
Max. cyl. press. lb/sq.in.
Fuel consumption lb/bhp-hr.
Fuel quantity, lb/cycle
6¼ in. Hg boost press. 145° valve overlap
8¾ in. Hg boost press. normal valve timing- 15° overlap
No boost normal valve timing

Fig. 10—Effect of Scavenging on the Quiescent-Combustion-Chamber Engine-Performance; Variable Fuel Quantity; 1500 R.P.M.; Net Results

Table 1—Nozzle Characteristics and Corresponding Engine Performance: Engine Speed, 1500 R.P.M.

Orifice Arrangement	Nozzle Designation, Number	Total Orifice Area, Sq. In.	Orifice Diameter, In.	Percentage of Total Orifice Area	Indicated Mean Effective Pressure, Lb. per Sq. In.		Indicated Fuel Consumption Lb. per I. Hp.-Hr.		Maximum Cylinder Pressure, Lb. per Sq. In.	Percentage Full-Load Fuel with Clear Exhaust
					¾ Full Load	Full Load	¾ Full Load	Full Load		
Miscellaneous Series; Low Rate of Injection for This Series										
	3	0.00013	A = 0.008 B = 0.006 C = 0.004	A = 38.1 B = 21.4 C = 9.5	78	Orifice Area Too Small	0.52	.....	650	.....
	4	0.00014	B = 0.008 C = .005	B = 35.9 C = 14.1	80		0.52	.....	650	.....
	7	0.00019	A = 0.008 B = 0.007 C = 0.005 D = 0.004	A = 26.4 B = 20.25 C = 10.3 D = 6.67	87		0.48	.....	650	.....
7-Orifice Series										
	9	0.00069	A = 0.007 B = 0.018 C = 0.005 D = 0.008	A = 5.58 B = 36.8 C = 2.84 D = 7.25	113	125	0.380	0.450	575	66
	10	0.00053	A = 0.007 B = 0.015 C = 0.005 D = 0.008	A = 7.26 B = 33.2 C = 3.7 D = 9.42	100	119	0.430	0.447	550	64
	11	0.00041	A = 0.007 B = 0.012 C = 0.005 D = 0.008	A = 9.52 B = 28.0 C = 4.85 D = 12.3	97	110	0.435	0.520	525	62
	12	0.00034	A = 0.007 B = 0.010 C = 0.005 D = 0.008	A = 11.5 B = 23.5 C = 5.08 D = 14.9	82	99	0.460	0.570	500	58
	14	0.00087	A = 0.007 B = 0.021 C = 0.005 D = 0.008	A = 4.43 B = 39.75 C = 2.25 D = 5.75	106	120	0.405	0.465	575	66
	16	0.00064	A = 0.007 B = 0.018 C = 0.005 D = 0.006	A = 6.0 B = 39.7 C = 3.06 D = 4.42	97	111	0.460	0.500	490	.....
	16-1	0.00069	A = 0.007 B = 0.018 C = 0.005 D = 0.008	A = 5.58 B = 36.8 C = 2.84 D = 7.25	104	117	0.420	0.480	510	.....
	16-2	0.00072	A = 0.007 B = 0.018 C = 0.007 D = 0.008	A = 5.18 B = 35.1 C = 5.32 D = 6.9	110	123	0.370	0.455	535	68
	17	0.00074	A = 0.007 B = 0.018 C = 0.005 D = 0.010	A = 5.19 B = 34.25 C = 2.64 D = 10.6	111	124	0.370	0.460	550	81
	17-1	0.00081	A = 0.007 B = 0.018 C = 0.005 D = 0.012	A = 4.75 B = 31.25 C = 2.42 D = 23.95	112	124	0.360	0.445	550	71
	18	0.00093	A = 0.007 B = 0.018 C = 0.012 D = 0.010	A = 4.14 B = 27.30 C = 12.15 D = 8.45	95	111	0.440	0.500	510	72
	18-1	0.00097	A = 0.010 B = 0.018 C = 0.012 D = 0.010	A = 8.10 B = 26.2 C = 12.65 D = 8.10	95	111	0.440	0.500	510	72
	18-2	0.00107	A = 0.012 B = 0.018 C = 0.012 D = 0.012	A = 10.5 B = 23.7 C = 10.5 D = 10.5	104	112	0.400	0.500	475	67
9-Orifice Series										
	C	0.00084	A = B = 0.018 C = 0.010 D = 0.008 E = 0.007	A = B = 30.2 C = 9.35 D = 5.98 E = 4.58	104	118	0.410	0.475	550	74
	C-1	0.00088	A = 0.007 B = 0.018 C = 0.010 D = 0.008 E = 0.007	A = 4.37 B = 28.8 C = 8.91 D = 5.69 E = 4.37	112	125	0.360	0.460	540	74

Orifice Arrangement	Nozzle Designation, Number	Total Orifice Area, Sq. In.	Orifice Diameter, In.	Percentage of Total Orifice, Area	Indicated Mean Effective Pressure, Lb. per Sq. In.		Indicated Fuel Consumption Lb. per I. Hp.-Hr.		Maximum Cylinder Pressure, Lb. per Sq. In.	Percentage Full-Load Fuel with Clear Exhaust
					¾ Full Load	Full Load	¾ Full Load	Full Load		
9-Orifice Series (Continued)										
	C-2	0.00092	A = 0.010 B = 0.018 C = 0.010 D = 0.008 E = 0.007	A = 8.53 B = 27.60 C = 8.52 D = 5.44 E = 4.18	111	125	0.380	0.450	550	73
	C-3	0.00078	A = 0.007 B = 0.018 C = 0.008 D = 0.007 E = 0.006	A = 4.93 B = 32.50 C = 6.40 D = 4.94 E = 3.63	102	122	0.415	0.460	500	71
E—Series										
	E_2	0.00051	B = 0.018	B = 50.	97	99	0.440	0.570	530	55
	E_4	0.00067	B = 0.018 C = 0.010	B = 37.9 C = 11.7	106	120	0.410	0.470	570	73
	E_6	0.00072	B = 0.018 C = 0.010 D = 0.006	B = 35.25 C = 10.9 D = 3.92	114	127	0.380	0.450	550	67
	E_8	0.00080	E_6 + 2—0.007		110	124	0.380	0.440	580	62
	E_{10}	0.00084	E_8 + 2—0.005		110	126	0.390	0.450	590	68
	E_{12}	0.00088	E_{10} + 2—0.005		106	119	0.400	0.470	570	65
	E_{14}	0.00092	E_{12} + 2—0.005		100	113	0.420	0.490	550	66
	E_{16}	0.00096	E_{14} + 2—0.005		100	113	0.430	0.500	570	71
	E_5	0.00066	A = 0.018 B = 0.014 C = 0.008	A = 38.4 B = 23.6 C = 7.5	114	123	0.370	0.460	550	75
F—Series										
	F_{20}	0.00072	B = 0.018 C = 0.010 D = 0.006	B = 35.25 C = 10.90 D = 3.92	114	127	0.380	0.450	550	73
	F_{23}	0.00072	B = 0.018 C = 0.010 D = 0.006	B = 35.25 C = 10.90 D = 3.92	114	124	0.365	0.460	560	75.5
	F_{26}	0.00072	B = 0.018 C = 0.010 D = 0.006	B = 35.25 C = 10.90 D = 3.92	114	124	0.380	0.460	580	80
	F_{29}	0.00072	B = 0.018 C = 0.010 D = 0.006	B = 35.25 C = 10.90 D = 3.92	114	124	0.375	0.460	580	74.5
H—Series										
	H_1	0.00089	B = 0.020 C = 0.011 D = 0.007	B = 35.10 C = 10.60 D = 4.30	119	128	0.360	0.440	600	73
	H_2	0.00100	B = 0.022 C = 0.011 D = 0.006	B = 38.00 C = 9.50 D = 2.83	115	128	0.360	0.450	600	70
	H_3	0.00083	B = 0.020 C = 0.010 D = 0.0055	B = 37.80 C = 9.42 D = 2.85	117	124	0.360	0.450	600	72
	H_4	0.00119	B = 0.024 C = 0.012 D = 0.0065	B = 37.80 C = 9.45 D = 2.77	114	128	0.365	0.445	590	64
	K_4	0.00097	B = 0.019 C = 0.014 D = 0.008	B = 29.1 C = 15.8 D = 5.2	123.5	136	0.340	0.410	880 C.R. = 15.3	73.7
	M_1	0.00087	A = 0.016 B = 0.011 C = 0.007	A = 23.10 B = 10.90 C = 4.42	120	136	0.343	0.400	770 C.R. = 15.3	70.7
	M_2		Two Outside A's Increased to 0.018		122	134	0.350	0.415	810 C.R. = 15.3	69

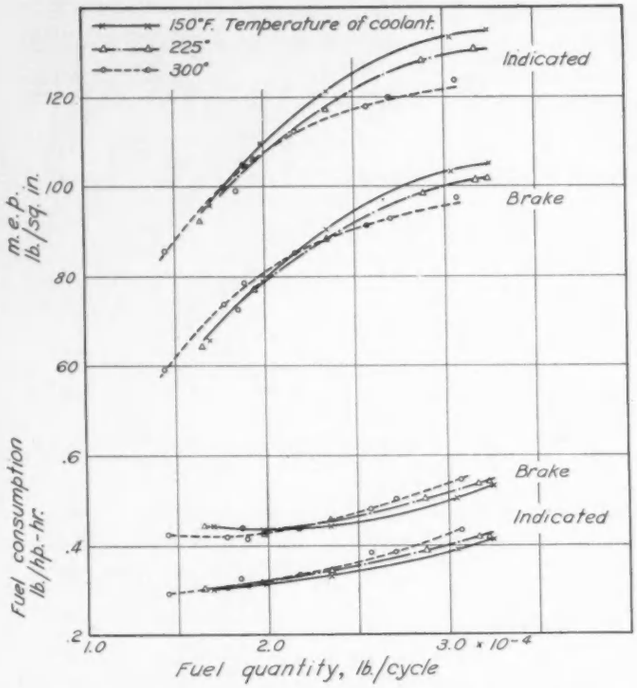


Fig. 11—Effect of Coolant Temperature on Quiescent-Combustion-Chamber Engine-Performance; 1500 R.P.M.

cylinder head by using a large valve-overlap to scavenge the chamber. Fig. 10 shows the improvement in net brake performance obtained by valve overlap and boosting. It should be noted that the scavenged performance with 6¼ in. of mercury boost is superior to the unscavenged performance at either 0 or 8¾ in. of mercury scavenging pressure at all fuel quantities.

The tests further showed that, without boost, the use of 145-deg. valve-overlap decreased the weight of air received by the engine but increased the brake mean effective pressure obtainable with clear exhaust and decreased the specific fuel consumption, demonstrating that removal of the residual gases improves the combustion characteristics. The largest improvement in clear-exhaust performance was obtained with a boost pressure of 2½ in. of mercury, from which the conclusion was drawn that with this boost pressure the combustion chamber was practically completely scavenged of residual gases and any further increase in performance was due to the additional air that could be trapped in the cylinder as the induction pressure was increased.

Coolant Temperature.—The effect of increased coolant-temperatures on the performance of this combustion chamber was investigated. Fig. 11 shows that performance was adversely affected except at light loads, where a coolant temperature of 300 deg. fahr. nets a slight improvement, due to reduced engine friction. At full load the 300-deg. fahr. coolant-temperature decreased the performance 8 per cent below that obtained with normal temperatures. Of this loss, 4 per cent can be attributed to decreased volumetric efficiency; the remainder to inferior combustion characteristics, probably to too short an ignition lag, destroying the distribution of the fuel and air.

Boosting the engine at high coolant temperatures showed no advantage to be gained by the combination.

Conclusions.—(1) To obtain optimum performance from the quiescent combustion chamber it is necessary to proportion and direct the fuel spray so that the fuel is distributed as uniformly as possible throughout the air. Impingement of the fuel spray on the piston and chamber walls is not neces-

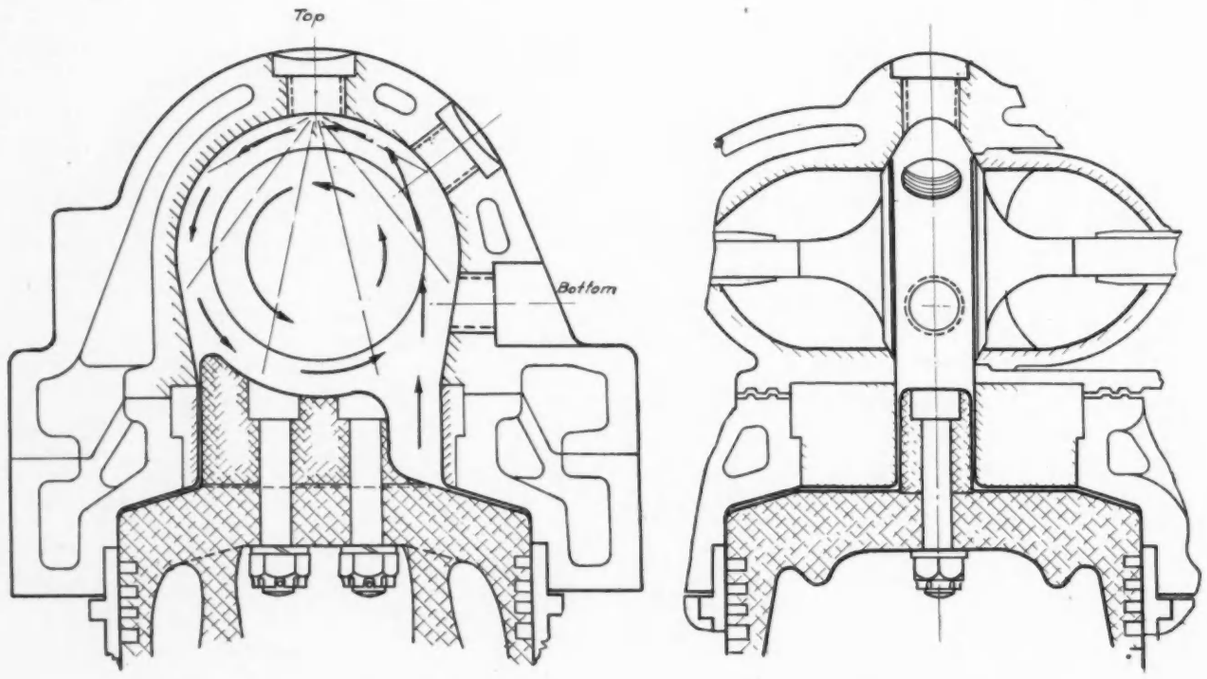


Fig. 12—Single-Passage Displacer Combustion Chamber; End and Side Views

sarily detrimental to combustion but may be an aid to distribution.

(2) The engine performance obtained by proportioning the areas of the orifices to the volume of air to be served by each orifice is approximately the same as that obtained by varying all orifice sizes and determining from the engine power the optimum combination. Neither method is complete in itself; the use of both together with a study of spray photographs should yield the best results.

(3) There appears to be no sharply defined optimum value for the number, direction, or size of the injection orifices. A six-orifice nozzle gave power and economy equal to that obtained with any other number and arrangement of orifices. Two injection valves connected to the same pump gave inferior performance.

(4) Boosting merely extends the performance curves and does not essentially alter their characteristics; however, it tends to make operation smoother.

(5) Coolant temperatures of 300 deg. fahr. decreased normal performance by about 8 per cent, of which 4 per cent was due to lower volumetric efficiency. Operation was smoother at the high temperature.

(6) The use of large valve-overlap allowed the vertical-disc chamber to be scavenged and outputs to be obtained that differed very little from those of the pre-chamber type, with the added advantages of easier starting and less combustion knock. Neither chamber gives outputs high enough to be useful in an airplane powerplant.

Displacer-Piston Type of Combustion Chamber

Another distinctive type of combustion chamber was created by setting up a forced airflow in the previously quiescent cylinder head and utilizing both directed airflow and directed fuel sprays to get the air and fuel together. This result was accomplished by bolting a lug or "displacer" on the piston crown to enter the cylinder head throat with mechanical clearance on the sides and form a restricted flow passage between one end and the throat wall as the piston approached top center. A forced airflow of varying velocity was thus created just before injection of the fuel charge. By a proper shaping of the top of the displacer, a symmetrical combustion chamber was formed to conserve the airflow. Fig. 12 shows the displacer-piston combustion-chamber.

Preliminary tests showed performance with this combustion chamber so far superior to that obtained from either the pre-chamber or the quiescent cylinder heads that comprehensive tests were started to determine the optimum passage width, displacer height, and fuel-spray characteristics for maximum performance. It soon became evident, however, that the displacer was overheating. This difficulty was overcome by casting the displacer integral with the piston crown. The restricted-flow area was then divided into two equal passages at either end of the rectangular displacer. Fig. 13 shows diagrammatically the combustion chamber thus formed, the probable airflow directions at various piston positions, and the fuel-spray directions and relative penetrations. It is believed that the airflow with the double-passage displacer was less orderly than that obtained with the single passage and set more of the air volume in the combustion chamber in turbulent motion. At any rate, the performance with the double-passage displacer was a little superior to that of the single-passage type. Consequently, all discussion to follow will concern the double-passage type. The compression ratio was 15.3:1.

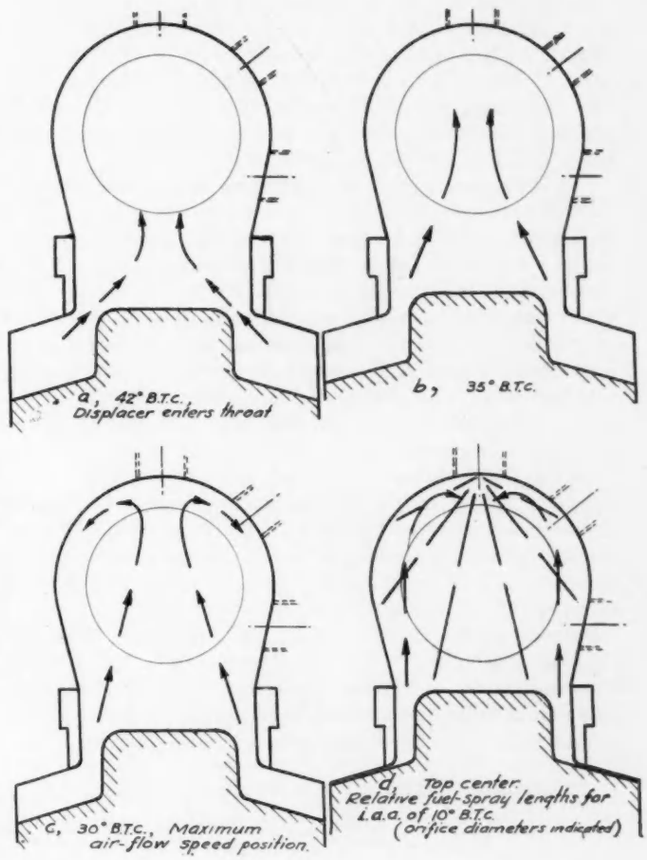

a, 42° B.T.C.
Displacer enters throat
b, 35° B.T.C.
c, 30° B.T.C., Maximum air-flow speed position
d Top center.
Relative fuel-spray lengths for I.a.o. of 10° B.T.C.
(orifice diameters indicated)

Fig. 13—Diagrammatic Representation of the Two-Passage Displacer Airflow and Fuel Sprays

Passage Width.—With a displacer height chosen to enter the cylinder head throat 42 deg. before top dead-center, tests were made to determine the optimum passage width for maximum performance. Fig. 14 shows the effect of various passage widths on brake mean effective pressures obtainable with clear exhaust, corresponding specific fuel consumptions, maximum air velocities (calculated) through the restrictions and friction mean effective pressure. A passage width of 19/64 in. is obviously optimum, which gives a calculated maximum velocity of forced airflow of eight times the linear speed of the crankpin.

Displacer Height and Shape.—Using the optimum passage width, the displacer height was varied, retaining the rectangular shape. Performance was not critical to height of displacer over a wide range, but was optimum for a height of 1 3/16 in., which allowed the restricting of the throat to start at 42 deg. before top dead-center, the height, oddly enough, that had been arbitrarily chosen for preliminary tests.

The direction of forced airflow and the timing of the maximum velocity attained were varied by sloping the ends of the displacer to form a trapezoidal shape in elevation approaching a triangle as the limit. Performance became, however, progressively worse with departure from the rectangular shape although the worst shape still gave performance superior to the quiescent combustion chamber.

Spray Distribution.—A limited number of tests were run

to determine the best fuel-spray arrangement to use in conjunction with the forced airflow. Although few in number, the nozzles tested covered the range of variation that previous experience had indicated to be most likely to affect performance. The conclusion was reached that the nozzle giving best performance with the quiescent combustion chamber of the same compression ratio was best for optimum performance when the displacer piston was used.

Fig. 15 gives a comparison of engine performance obtained with this cylinder head without and with the airflow created by the displacer and shows the decided superiority of the latter. The brake mean effective pressure obtainable with clear exhaust is increased to 115 lb. per sq. in. by the use of the airflow, corresponding to clear exhaust with but 15 per cent excess air, a value which, so far as is known, has not been obtained from a high-speed compression-ignition engine with any other type of combustion chamber. Best fuel economy is seen to be 0.41 lb. per b.hp-hr. Maximum indicated mean effective pressure of 165 lb. per sq. in. is obtained with smoky exhaust. More will be said about friction later.

A comparison of the rates-of-burning curves of Fig. 16 shows the superior performance of the displacer-piston combustion-chamber to result from earlier burning of the fuel in the cycle, as well as utilization of more of the available air, allowing the realization of a greater expansion ratio.

Boosting.—The aircraft engine will always be concerned with any means that increases the power output, one of which is the utilization of a greater air-charge. The effect on performance of boosting the displacer-piston combustion-chamber is shown in Fig. 17. Maximum cylinder pressures were

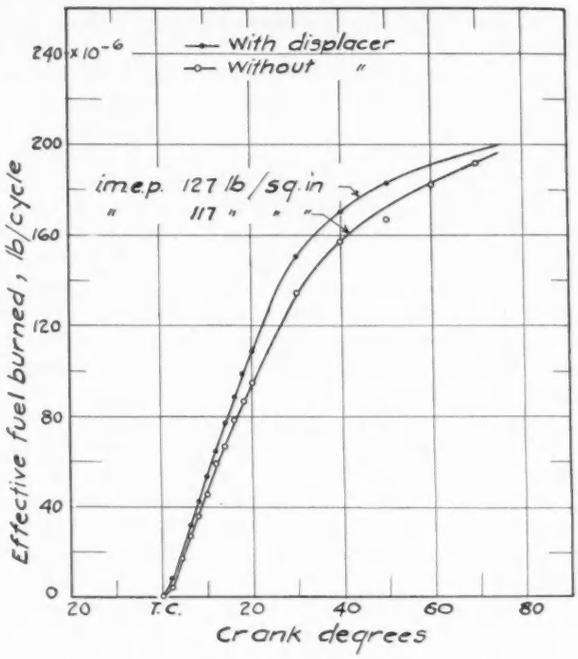


Fig. 16—Comparison of Rates of Burning With and Without Displacer; 1500 R.P.M.

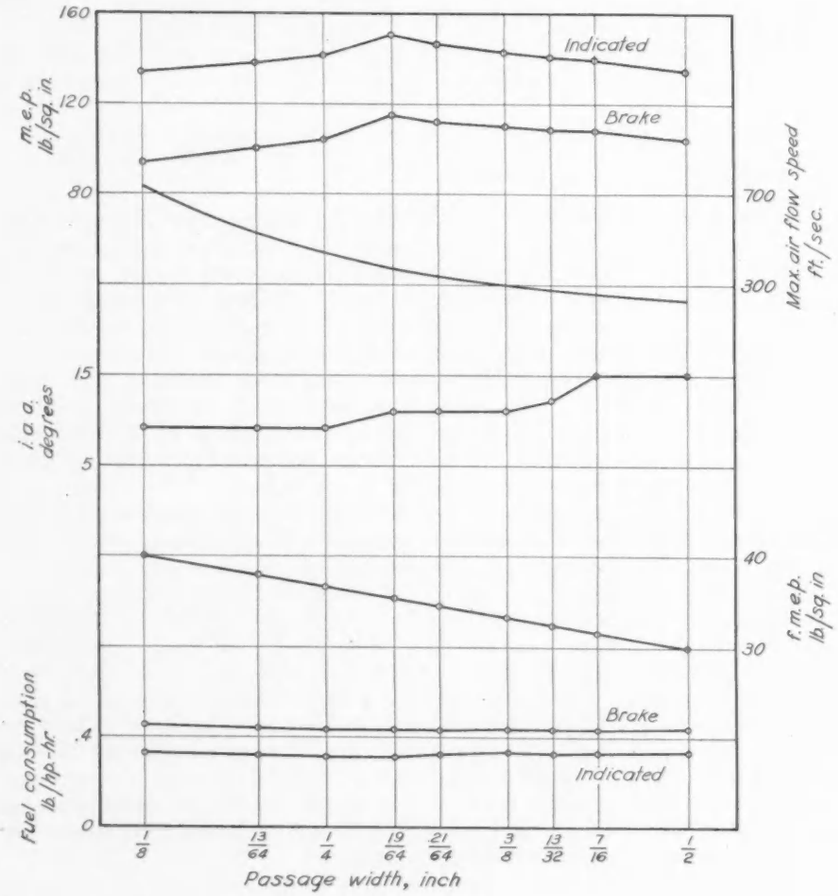


Fig. 14—Effect of Passage Size on Engine Performance; Double-Passage Displacer; 1500 R.P.M. Clear Exhaust

limited to 900 lb. per sq. in., which necessitated retarding the injection advance as the boost pressure increased and detracted from the specific performance. In general, the effect of boosting, as found with the other combustion chambers, is to extend the load range of the engine, the net performance at light loads and high boost suffering from the power required to supply the excess air under pressure. At 10 in. of mercury boost, a maximum net brake mean effective pressure of 160 lb. per sq. in. was obtained, corresponding to an indicated mean effective pressure of over 210 lb. per sq. in. No improvement in maximum net clear-exhaust output is obtained by exceeding 7½ in. of mercury boost, where 131 lb. per sq. in. brake mean effective pressure was obtained, indicating the need for further study of the optimum conditions required with boosting.

High-Temperature Coolant.—Brief tests were made with the displacer-piston combustion-chamber, both boosted and unboosted, using coolant temperatures as high as 280 deg. fahr. The decrease in performance was not so great as that found with similar tests with the quiescent combustion chamber (4 per cent compared with 8 per cent). Furthermore, the displacer-piston combustion chamber showed inferior performance at all loads with high-temperature coolant; whereas the quiescent chamber showed improvement at fuel quantities below 0.000225 lb. per cycle. From these tests it appears that a decrease in ignition lag does not affect the combustion adversely to the same or as great an extent as without airflow.

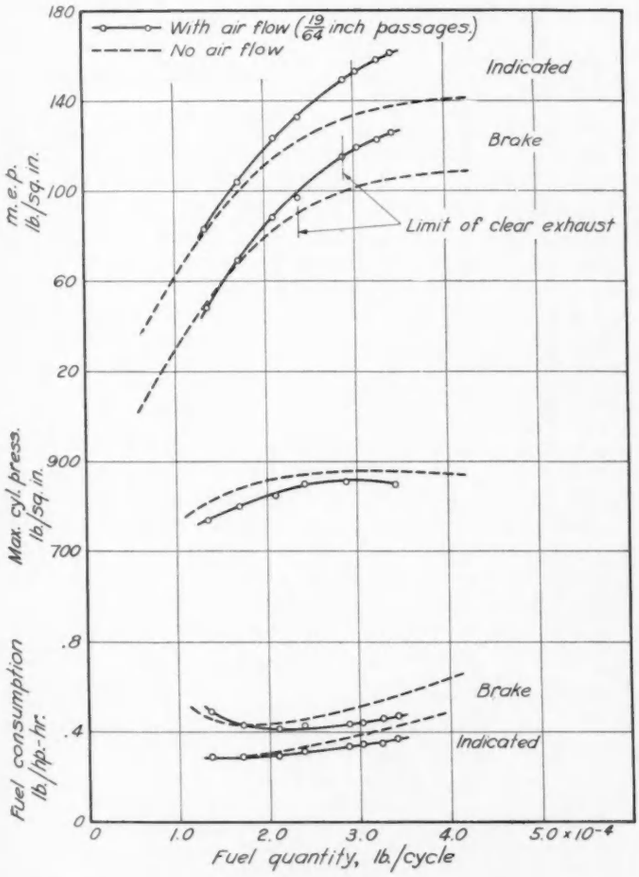


Fig. 15—Comparison of Engine Performance With and Without Airflow; Double-Passage Displacer; 1500 R.P.M.

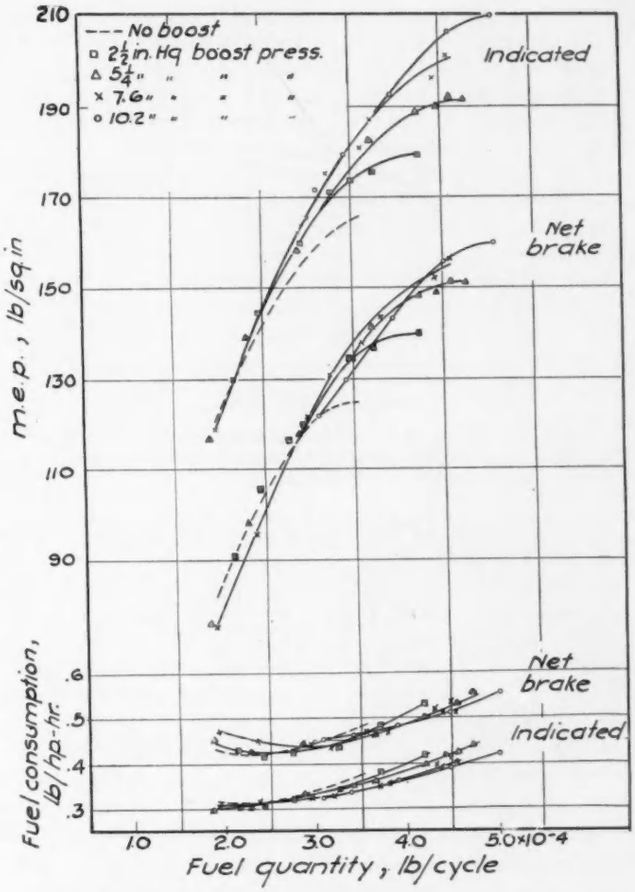


Fig. 17—Effect of Boost Pressure on Displacer Engine Performance at 1500 R.P.M.

Friction—Considerable data have been presented, inspection of which shows the friction of the test engines to be very high. The result of high friction is to detract from the brake output and economy. Results of a short series of tests supported the belief that the single-cylinder test-engine friction is inordinately high compared with that to be expected in a multi-cylinder engine. Fig. 18 is presented to show the effect of compression ratio on friction mean effective pressure of an integral combustion chamber (no airflow restrictions) when motored at 900, 1200, 1500 and 1800 r.p.m. Other tests were conducted to show the effect of oil temperature, coolant temperature, pre-chamber clearance distribution, and analysis of loss occasioned by the various component parts of the engine. This work will be shortly made available by the Committee.

The most striking point about the results of Fig. 18 is the high friction mean effective pressures at carburetor-engine compression-ratios. The friction mean effective pressure at a 6:1 compression ratio is 50 per cent greater than that found in multi-cylinder carburetor-engines. It seems reasonable to assume, therefore, that multi-cylinder compression-ignition-engine friction will be likewise considerably less than that of the test engines and that net brake outputs and fuel consumption herein presented will be correspondingly improved.

On this basis, a propeller-load curve (Fig. 19) has been prepared comparing a commercial carburetor aircraft engine and a hypothetical multi-cylinder displacer-piston-type compression-ignition engine of the same displacement. By boosting the latter to 10 in. of mercury—which is considered allowable,

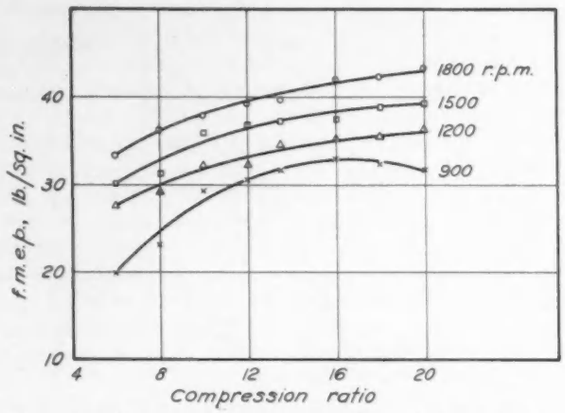


Fig. 18—Effect of Compression Ratio on Friction Mean Effective Pressure; Integral Combustion Chamber; 1500 R.P.M.

since the amount of boost is not limited by detonation as in the spark-ignition engine—and assuming a mechanical efficiency of 81 per cent attainable at 2000 r.p.m. engine speed, the same power output is given by the two engines at a substantial advantage to the compression-ignition engine in fuel economy. Clear exhaust will be obtained at speeds up to 1900 r.p.m. It is believed that this favorable comparison to the compression-ignition engine is conservative, inasmuch as further development of boosted performance should yield substantial improvement.

Summary

It is concluded that the compression-ignition-engine performance has improved steadily to the point where the gasoline engine has a serious competitor in performance and economy. The increase in compression-ignition-engine performance has been directly caused by better methods of mixing the fuel and the air. Further improvements in performance may be expected by further improvements in fuel and air mixing. Optimum mixing will be obtained with a proper combination of fuel spray and airflow. The displacer-piston combustion-chamber, although the best so far developed, is not to be considered the ultimate solution. Analysis shows the possibility of a 27.5-per cent increase in performance over the best thus far obtained.

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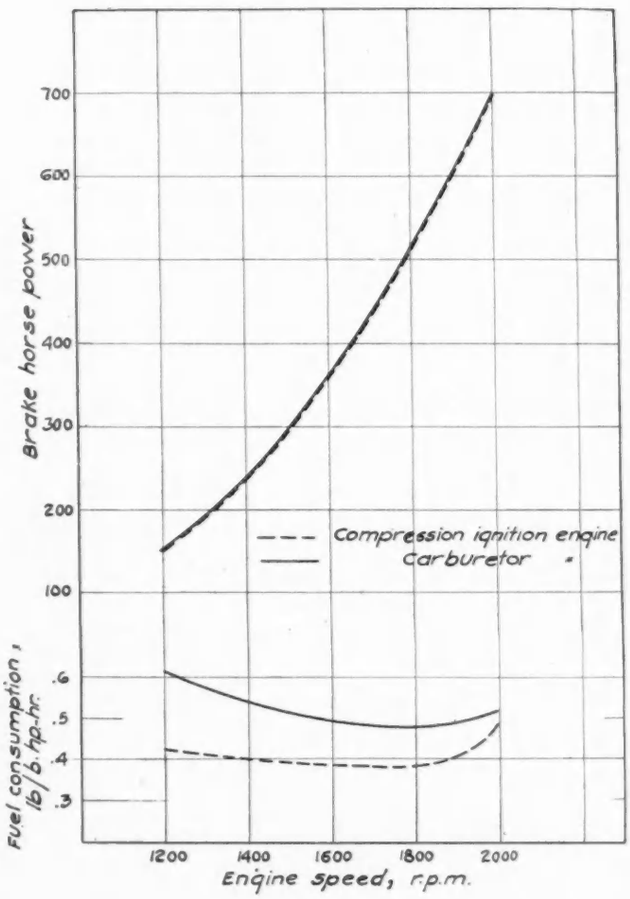

Fig. 19—Comparison of Propeller Load Performance for Carburetor and Compression-Ignition Engines of 1690-Cu. In. Piston-Displacement

Maximum boost pressure for the carburetor engine, 4 in. of mercury; for the compression-ignition engine, 10 in. of mercury

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Discussion

Questions Asked on Ignition Quality and Cylinder Wear

—**G. H. Freyermuth**
Standard Oil Co. (N.J.)

WE wish to congratulate Mr. Whitney and the N.A.C.A. on the excellent nature of the experimental work disclosed in this paper. It seems to us that it is largely by means such as this that the parallel problems of engine performance and fuel quality will ultimately be solved.

A question, perhaps somewhat premature, naturally arises to any fuel manufacturer in connection with this work. We have all tacitly assumed that, as efficiency and performance of the compression-ignition engine are improved, fuel-ignition-quality requirements are lessened. Has Mr. Whitney been able to determine any definite indications of the effect of combustion-chamber efficiency on required ignition quality of the fuel and, if not, is work such as this contemplated in further development of the problem?

We note the conclusions in Mr. Whitney's paper that, as coolant temperatures are decreased, engine efficiency is improved. It is to be hoped, however, that this does not presage a general movement toward low jacket-temperatures without further investigation of the resultant effect on cylinder wear.

Variations in Ignition Lag Prevent a General Conclusion

—**Ernest G. Whitney**
National Advisory Committee for Aeronautics

A GENERAL conclusion cannot be drawn from the data now at hand as to the effect of engine performance and efficiency on fuel-ignition quality requirements. The ignition lag of a given fuel is influenced by factors that may be characteristic of engines of either good or poor performance. For example, the Standard Oil Company's L-1 reference fuel used in the three combustion chambers discussed showed ignition lags under comparable conditions of about 7 crankshaft degrees at 1500 engine r.p.m. for the pre-chamber head, 14 deg. for the quiescent head, and 12½ deg. for the displacer head. Ignition lag is shortened by increased compression ratio, increased cylinder temperature, increased airflow, and by boosting. The ultimate combination of factors for the optimum combustion chamber remains a matter of conjecture. More definite trends may be revealed by results of our projected study of injection rates and requirements.

Diesel and Otto Cycle Differences

THE Diesel-cycle principle is based on a law of nature that the more air is compressed the higher the temperature of this air becomes. Dry air with an original temperature of 60 deg. fahr., if compressed into one-tenth of its original volume without heat loss through cylinder walls, will reach a temperature of approximately 825 deg. fahr.; if compressed into one-fifteenth of its original volume it will be approximately 1050 deg. fahr. and so on; that is, the greater the compression, the higher the temperature. The compression-ignition engine makes use of this means to ignite the fuel instead of using an electric spark as in a conventional gasoline engine.

In extremely cold weather it may be necessary to apply artificial heat to the intake manifold or to heat the circulating water in the jackets or the lubricating oil in the base to accomplish quick starting. This is due to the heat generated in the air that is being compressed radiating rapidly into the cylinder wall.

When the cylinder walls are very cold, such as in zero weather, they may absorb so much of this heat from the air that the air temperature may be below the ignition temperature of the fuel oil at the time the oil is injected.

By using fuel-injection pumps and nozzles, fuel oil can be used for fuel instead of gasoline. The pumps raise the fuel to very high pressures and force it through a relatively small opening, or openings, in the nozzles; this atomizes or breaks the fuel into very fine particles in the form of mist and sprays this mist into the hot compressed air where it ignites and burns.

Any oil, when subjected to a certain temperature, will ignite and burn. Various Diesel-fuel oils have different ignition temperatures, but practically all are well below 1050 deg. fahr.

The basic difference between the Diesel-cycle engine and gasoline Otto-cycle engine is that, in the Diesel cycle, the heat for firing the fuel is in the air in the cylinder previous to the introduction of fuel and, in a gasoline Otto-cycle engine, the fuel is in the air of the cylinder previous to the introduction of heat—the spark—to fire the fuel.

As fuel oil is heavier than gasoline and much harder to vaporize due to its natural formation, it must be vaporized in a mechanical way instead of exposing it to rapidly moving air. Basically both engines are similar otherwise.

Excerpt from a Canadian Section paper on "High-Speed Diesel-Engine Application to Road-Transport Vehicles", by E. J. Cosford, president, Associated Equipment Co. of Canada, Ltd., Quebec.

Aircraft Spark-Ignition Versus Compression-Ignition Engines

By Kenneth A. Browne
Project Engineer, Wright Aeronautical Corp.

THE need for improvement of fuel consumption in modern aircraft is stressed. The basic economy of oil engines for airline service is briefly summarized, together with the effect of lower fuel consumption on the range and payload of airplanes.

The theoretical efficiencies of the Otto and Diesel cycles are compared with the efficiency actually obtained on present aircraft engines. The data are presented in condensed graphic form for easy comparison. A prediction is made of the fuel economy that may be expected from the gasoline aircraft-engine in the near future, together with a summary of the means required to obtain it.

The prospective place of the compression-ignition engine in commercial and military service is briefly outlined. An analysis of the weight possibilities of compression-ignition engines as compared to present gasoline engines is made. The two-stroke engine is selected as the type worthy of further development, and indications are that it will fall into the weight class with present gasoline engines. Both engine types will be further developed, and a specification for the future competitive Diesel is outlined.

DURING the last decade, tremendous strides have been taken in the improvement of the performance and reliability of aircraft. Cruising speeds and range of large craft have approximately doubled. The engine manufacturer has contributed engines of greater power and increased reliability, but the aircraft manufacturer has contributed the improvement in efficiency which accounts for a major part of the increased performance. The parasitic drag of the best airplane has been reduced to a figure so low that its entire elimination would produce an increase in performance of but a few per cent.

Variable-pitch propellers are no longer in the experimental stage, and have added considerably to the performance of up-to-date aircraft. The ultimate in this direction seems to be the constant-speed propeller, which is but a variable-pitch propeller provided with an automatic pitch-controlling mechanism designed to maintain constant propeller speed regardless of the engine-throttle opening. Manual adjustment of the governed speed allows maximum propeller efficiency to be obtained at the various flight conditions. Engine efficiency, as expressed by specific fuel consumption, has improved but little. The familiar large air-cooled radial-engine of today is the result of literally millions of dollars spent on experimental development; but, until 1934, its operating thermal efficiency has been less than that of many old war-time water-cooled engines. This last year has seen some real improvement in fuel consumption; not the result of better fuel, but of improved design. There remains, however, a margin for further improvement, particularly in the service utilization of the engine's possibilities.

Economy of the Fuel-Oil Engine

Usually, comments on the desirability of using fuel oil in compression-ignition engines to lower operating costs are considered out of order because the fuel cost is supposed to be insignificant. This attitude is a consequence of bus and truck operating-figures. However, let us consider the figures on U. S. airline operations for 1934 as presented in the Transport Report of the Bureau of Air Commerce of the U. S. Department of Commerce:

Number of Passengers	537,637
Miles Flown	42,622,619
Gasoline Used, gal.	21,991,782

The average number of miles per gallon of gasoline is 1.94, a figure slightly lower than shown in 1932 and 1933 and probably the result of faster schedules. Passenger miles per gallon of gasoline are given at 7.23 which, figuring fuel cost at 11 cents per gal., means 1.52 cents per passenger mile, a remarkably high figure. The average airline-passenger fare in the U. S. in 1934 was close to 6 cents per mile, which places the fuel cost at 25 per cent of the passenger revenue. If the fuel cost is insignificant to the total operating cost, it is easy to understand why airlines claim to need mail subsidies for profitable operation. However, as passenger traffic on the airlines increases, schedules speed up, and fares reduce, the fuel bill will become a still more important item. Regardless of engine capabilities, the average specific fuel consumption now obtained on modern airline operation is in excess of 0.52

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 17, 1935.]

lb. per hp-hr. If the airlines had been equipped in 1934 with compression-ignition engines using oil at 4 cents per gal. and cruising at 0.36 lb. of fuel per hp-hr., the consumption of a Junkers Diesel, instead of the optimistic average figure of 0.53 for gasoline at 11 cents, the cost per passenger mile would have been 0.32 instead of 1.52 cents. Further indulgence in this assumption shows a saving in operating cost of 20 per cent of the passenger revenue, and a total economy to the airlines in the neighborhood of $1,910,000; by no means an insignificant figure.

The transoceanic airliner, a mere dream in 1920, is today a reality. The new Martin Model-130,—a 50-place flying-boat for the Pacific service of the Pan American Airline—is designed for a cruising range of 4000 miles with considerably less than a 50-passenger payload, and the engines must cruise at a fuel consumption of about 0.45. By constant checking and controlling, this fuel consumption may be maintained. Again, assuming these ships to be equipped with compression-ignition engines which would cruise at 0.36 lb. per hp-hr., the weight of fuel saved in 3000 miles would be about 3500 lb., which represents a corresponding increase in payload or allows 17 extra passengers. In the instance of operation with the designed payload, the range of these ships would be increased to 5000 miles by the use of the assumed oil engines. Of course these performance figures do not apply if the gasoline engine were developed to produce power nearly as economically as the oil engine; further, they are based on equal engine weights per horsepower.

Theoretical Limitations of Efficiency

Any discussion of efficiency of internal-combustion engines necessarily involves a reference to the thermal efficiency of the ideal cycles, technically known as the Otto and Diesel, so named from the men who first investigated them. Strictly speaking, our high-speed engines do not operate on either of these theoretical cycles. Combustion in the spark-ignition engines requires an interval of time and, therefore, heat is added to the working fluid only approximately at constant volume. The interval required for combustion, together with the effect of dissociation of the products of combustion, produce the rounded top of the familiar *P-V* diagram. With the solid-injection compression-ignition engine, a certain time-lag in the ignition of the liquid fuel occurs which necessitates relatively early injection at high-speed operation. Considerable fuel, therefore, is present in the air charge before combustion takes place, which results in a rapid pressure rise at approximately constant volume (Otto-cycle operation), followed by further combustion of the later part of the injected fuel at a reducing pressure during the beginning of the expansion stroke. The actual engines, therefore, operate on what is termed a mixed cycle. The cycles of so-called Diesel engines are often expressed as 60-per cent Diesel, 40-per cent Otto, or whatever percentages are found to exist. If much of the Otto-cycle characteristic is present, the engine has what is termed the "Diesel knock", another engineering misnomer, for it should be called the "Otto knock". (See Fig. 1)

The air-standard efficiency has for years been used as a reference for analyzing engines; primarily because of the simplicity of its mathematical expression. It assumes the working fluid to be air and the ratio of specific heats constant; a rather unattainable condition. The equation for efficiency is

$$e = 1 - (1/r)^n \qquad (1)$$

in which r is the compression ratio and $n = 0.4$, the ratio of specific heat at constant pressure to that at constant volume, less one. Calculation of the ideal cycle efficiency, taking into consideration the varying composition of the mixtures of actual working gases through the cycle, the variation in their specific heats, the equilibrium conditions of dissociation at high temperatures and the effect of continued combustion during the expansion stroke, is an extremely involved and laborious procedure. Fortunately, this work has been admirably done for both the cycles in question and presented in excellent form by Goodenough and Baker[1]. The essence of their work is expressed in three empirical formulas for the value (n) in the Equation (1). They are:

Otto cycle with mixtures leaner than required for complete combustion:

$$n = 0.3867 - \frac{6.5}{a - 35} - \frac{0.043}{r} \qquad (2)$$

Otto cycle with rich mixtures; excess fuel:

$$n = 0.524 - \frac{24.6}{a} \qquad (3)$$

Diesel cycle:

$$n = 0.434 - \frac{19.5}{a - r} - \frac{0.7}{r} \qquad (4)$$

in which a is the percentage of air supplied, based on 100 per cent for the chemically correct amount for complete combustion. These formulas were used to determine the ideal-efficiency curves shown in Figs. 2 and 3.

The compression-ratio scale of Fig. 2 was arbitrarily established to make the air-standard-efficiency curve for the Otto cycle a straight line. This results in a figure which is more useful visually and further emphasizes any departures of other curves from the relation of proportionality to the air standard. The ideal Otto-cycle efficiencies are shown for the two conditions of operation in which we are interested; namely, maximum-power and maximum-economy mixture-ratios. The two mixture ratios will vary somewhat on different engines but, for the sake of clarity, they have been assumed to fall within the range of 0.075 to 0.060 fuel-air ratio or 89 to 111 per cent excess air.

Unfortunately, the spark-ignition engine as we know it is inherently incapable of operating with high excess air because the extremely lean mixtures are not combustible from the electric spark; therefore, the ideal indicated efficiency for the spark-ignition engine falls within the shaded area. The ideal Diesel-cycle efficiency is shown for two mixture ratios, 0.0555

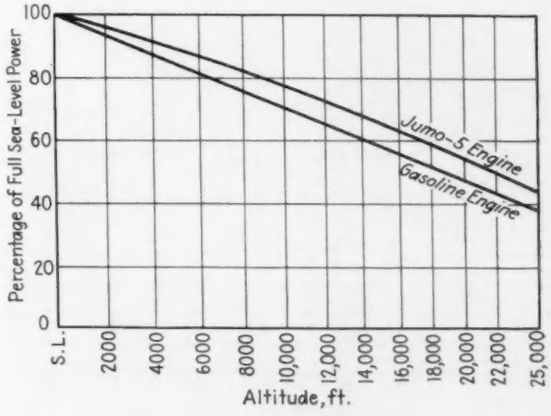

Fig. 1—Comparison of Full-Throttle Altitude-Performance of Compression-Ignition and Spark-Ignition Engines

[1] See Bulletin No. 160, Engineering Experiment Station, University of Illinois; G. A. Goodenough and John B. Baker.

and 0.0222, corresponding to excess-air figures of 120 and 250 per cent. These ratios have been selected on the basis that 120 per cent represents the condition at the practical maximum output of compression-ignition engines operating with clean combustion. Quite a few engineers will no doubt consider this figure an ideal toward which to strive, but at least one firm claims to operate engines at 110 per cent with clean exhaust. The 250-per cent excess-air figure represents a shot in the dark at a reasonable fuel-air ratio for an aircraft engine operating at maximum economy on cruising horsepower, about 70 per cent of the rated power. The shaded area in the high-compression range corresponds to the shaded area in the low-compression range and gives a visual impression of the ratio of the ideal efficiencies (about 40 to 50) of the spark-ignition and compression-ignition types. Further, the effect of fuel-air ratio on efficiency is seen to be greater than that of compression ratio, an effect which is seldom appreciated and is doubly apparent in Fig. 3.

The ideal curves in Fig. 3 are based on the same data used for Fig. 2, but express the limiting relations in a different way. Specific fuel consumptions in pounds per horsepower-hour are plotted against the fuel-air ratio. The data are based on 15 parts of air to 1 part of fuel by weight for 100 per cent complete combustion and a low heating value for the fuel of 19,500 B.t.u. per lb. As in Fig. 2, the abscissa scale is arbitrarily arranged to obtain straight lines for the ideal fuel-consumption curves. The shaded area in the low fuel-air-ratio region represents the ideal fuel consumption in the range of compression ratios used in compression-ignition practice; as does the shaded area on the rich end define the limits for the spark-ignition engine operating with practical compression ratios. For the convenience of those who are optimistic enough to consider a higher compression ratio, the dotted line for 10 to 1 is shown. Theoretically it shows but a 5 per cent gain over the 8.5 ratio.

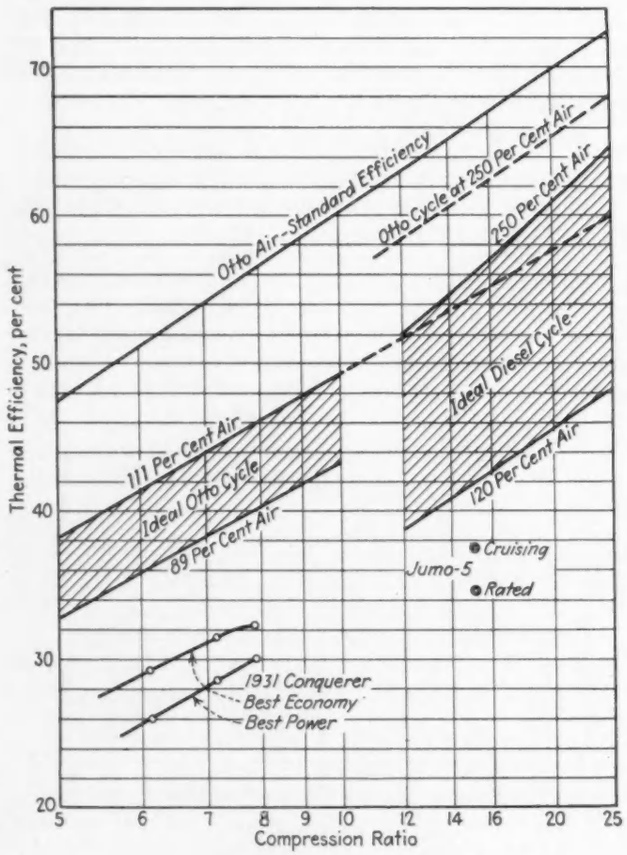

Fig. 2—Ideal and Actual Effects of Compression Ratio on Thermal Efficiency

H. R. Ricardo, in his classic on internal-combustion engines, shows curves of performance versus compression ratio which indicate no gain in brake thermal efficiency from 8.5 to 10 to 1, due to the increased frictional losses in an engine made heavy enough to stand the greatly increased peak pressures. Furthermore, petroleum fuels must be further developed to allow operation at the minimum fuel-air ratio for a compression ratio of 10. The efficiency of the Otto cycle is independent of the mean effective pressure, but the Diesel cycle is not independent. At no load, Diesel efficiency approaches the air-standard efficiency and steadily decreases as the mean effective pressure increases. This characteristic accounts for the difference in slope of the curves. The bottom of the ordinate scale is placed at 100 per cent thermal efficiency. The points on the zero fuel-air-ratio line are computed on the air-standard efficiency, and it is interesting to note how close they fall to the extension of the ideal cycles at zero output.

Experimental Data on Service Engines

In 1931, extensive tests were made on the Conqueror engine using three different compression ratios and operating with the coolant temperature at 180, 240 and 300 deg. fahr. For the benefit of those who are not familiar with this engine by name let it suffice to say that the Curtiss Conqueror is a liquid-cooled normally aspirated V-12 engine of 1570-cu. in. displacement. It was rated at 600 hp. at 2400 r.p.m., but the data shown in Figs. 2 and 3 for this engine are given in the neighborhood of 70 per cent of the rated horsepower. Further, it must be remembered that the data for this engine and the others mentioned in this paper were taken during carefully conducted laboratory experiments and therefore do not represent conditions actually obtained in flight service.

The two curves of Conqueror-engine performance shown on Fig. 2 are for brake thermal efficiency at best power-setting and at best economy with all other variables adjusted for the best performance. The fact that little improvement of maximum economy occurs between the 7.2 and 7.8 compression ratios is worth noting. The divergence from proportionality with the theoretical curves is the result of the fuel used during the test. Since that date fuels have been developed with higher octane rating. The 7.8 compression ratio can now be operated at a minimum consumption in the cruising range of 0.40, which corresponds to a brake thermal efficiency of 32.6 and this would considerably straighten out the line shown in Fig. 2. For the sake of comparison, two points representing the thermal efficiency of the Junkers Jumo-5 two-cycle compression-ignition engine are also plotted in Fig. 2. Data for compression ratios other than 14.9 to 1 are not available.

The results of recent experimental testing on the Wright Cyclone, an 1820 cu. in. single-row radial air-cooled engine, as well as recent results on the Conqueror engine, are shown in Fig. 3. The curve marked Cyclone-*G* 7000-ft. rating shows a very marked improvement over the upper curve marked Cyclone-*F* 7000-ft. rating, to which it compares in respect to horsepower and amount of supercharging even though the compression ratios of the two engines are almost identical. The lower fuel consumption is the result of numerous detail improvements in the Cyclone-*G* engine, the most important of which are marked reduction in cylinder temperature due to

improved cooling, a distinct improvement in spark plugs and a change in the manifolding. The *G* model not only can be experimentally tested at the low fuel consumptions but has proved to be capable of operating continuously at excellent economy. The same can hardly be said of the early models of so-called air-cooled engines, often sarcastically referred to as fuel-and-oil cooled; for, although they could be calibrated at fairly low fuel consumption, they would shortly disintegrate under continuous operation because of the elevated temperatures of the cylinder head and barrel. Again data for the Jumo-5 engine are shown for comparison with the theoretical Diesel cycle. The fuel-air ratio is estimated from the reported scavenging characteristic of the engine and, therefore, the points may not be located exactly; but it is believed they represent conditions accurately enough for this discussion.

Contemplated Fuel Economy for Gasoline Engines

The dotted curve at the bottom of the engine characteristics of Fig. 3 is the most optimistic fuel consumption that may be expected in the near future from the Cyclone-*G* model with 8 to 1 compression ratio, assuming of course a fuel of sufficiently high octane rating to allow operation at the low fuel-air ratios. It is estimated that a fuel in the neighborhood of 92 octane rating will be required to obtain the results, although possibly 87 octane rating might be sufficient for the satisfactory functioning at 70 per cent of rated power at rated altitude. At least a 100-octane fuel will be required for take-off conditions at the foregoing rated power.

Assuming for the moment, the optimistic minimum fuel consumption of 0.38 per hp-hr. for the cruising horsepower is obtained in the laboratory under careful control; it cannot be used or even approximately approached in flight service by the procedure and practices now employed. As previously stated, airlines are obtaining fuel consumptions no better than 0.53 lb. per hp-hr. and this from engines capable of operating at 0.48. Several fairly complicated pieces of apparatus must be installed and the flight personnel trained to use them before a high degree of fuel economy can be maintained in service.

First, a supercharger or boost regulator will be required to prevent accidental increases in the manifold pressure which would cause detonation on the low tetraethyl-lead-content fuel which is desired for cruising. Two fuels are not required for operation, but it is the common practice to arrange for a high-octane fuel for take-off and emergency and a lower lead-content fuel for cruising, in which case means should be provided to assure the switching to the high-octane fuel and the enrichening of the carburetor before the regulator is moved to a higher boost setting.

Second, an automatic mixture-control mechanism will be required for the carburetor to relieve the pilot of constantly checking the mixture strength. This device is another piece of complication added to an already involved mechanism, and is not complete in itself because manual adjustment of the automatic control will be essential to obtain best economy under varying atmospheric and operating conditions. In other words, the automatic control cannot be designed to control for maximum economy, but only for best power.

Third, because of the inaccuracies of mixture adjustment by the pilots, an exhaust-gas analyzer of some reliable design will be essential to give the pilot a positive indication of combustion conditions. The advent of the constant-speed propeller has removed the old customary "drop in revs" encountered when the mixture was too lean, and therefore introduced a problem in adjustment of the mixture strength which today requires an immediate solution if fuel economies even as good as heretofore are to be retained. And, when all this control is provided, a slight margin over maximum economy should be maintained which places the hypothetical cruising fuel consumption between 0.39 and 0.40 lb. per hp-hr.

Before passing to the compression-ignition engine, some recognition of the effect of the manifold fuel-injection system as developed by the U. S. Air Corps on fuel consumption must be considered. Improvement in fuel consumption on not-too-efficient carburetor-engines of 12 per cent has been obtained. Lack of data on an already highly efficient carburetor engine leads us to believe that considerably less than 12-per cent improvement would be obtained; for, our contemplated minimum consumption of 0.38 reduced by 12 per cent would result in a figure beyond our expectations. The use of the fuel-injection system, however, might be a help in obtaining the 0.38 figure for fuel consumption.

What Place Has The Oil Engine?

The usual difference in weight between the compression-ignition and the spark-ignition engine is ordinarily justified by the lower fuel consumption of the compression-ignition type. The weight advantage of the 2 lb. per hp. oil-engine, however, becomes apparent only on trips of over 5 hr. based on present gasoline-engine performance. Most of our airline schedules call for stops more frequently than every 5 hr., not necessarily to reload with fuel but to take on and discharge passengers; therefore, the major proportion of airline operators will have to be sold on features other than fuel consumption before adopting oil engines which weigh appreciably more than the present gasoline-engine equipment unless overall dollars-and-cents savings can be assured. Proposed trans-oceanic operations will present a different problem. The hue

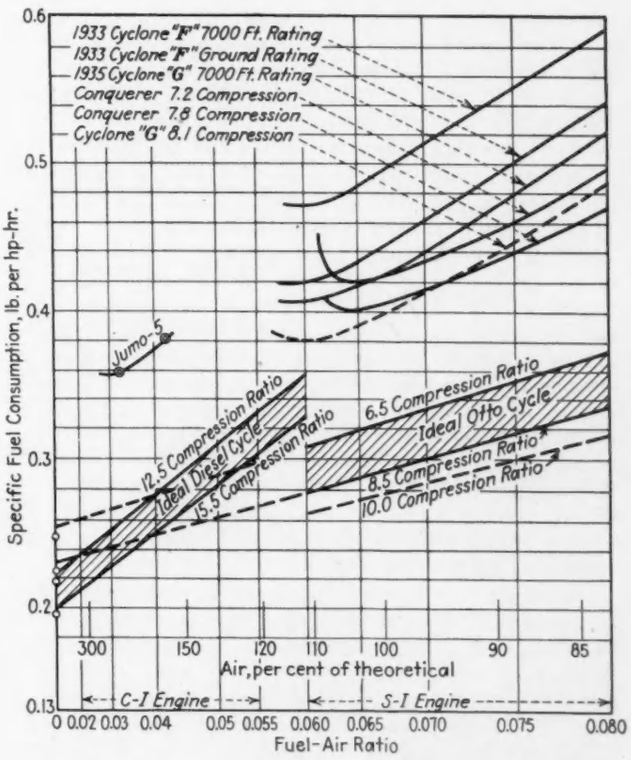

Fig. 3—Ideal and Actual Effects of Fuel-Air Ratio on Specific Fuel Consumption

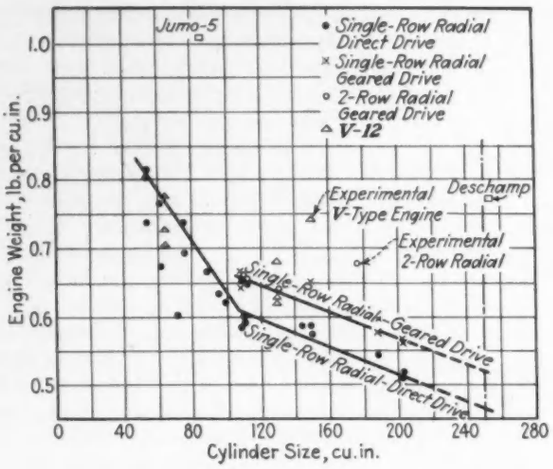

Fig. 4—Design Weights of Modern Aircraft Engines

and cry will be for lower and lower fuel consumption and, although the engine weight is less important on the long trips, the plane designers will request engines as light as the present models. If the gasoline engine can be made to cruise at 0.4 consumption, the weight benefit of the 2 lb. per hp. oil-engine is apparent at the end of 21 hr. and not 5 hr.; and 21 hr. will cover the distance from New York to Paris.

It would seem that the compression-ignition engine will not come into general aircraft use until it practically equals the gasoline engine in regard to weight and surpasses it in operating economy, with the possible exception of a service in which the reduction in fire hazard is of primary importance. Such an engine is not at all out of the realm of possibilities; and, if it should be developed and proved reliable, little resistance would be found to its general adoption. The simplicity of control of the C-I engine is an outstanding advantage. When the pilot throttles back to cruising power, the laboratory measured fuel economy is automatically obtained without further adjustments. The full power is instantly available for emergency without switching fuel tanks and, furthermore, full power will be obtained with little increase in specific fuel consumption, a feature quite valuable in case of the failure of one or more engines of a multi-engined ship. Further, elimination of the complicated pieces of control apparatus and instruments required by the low-consumption gasoline-engine will be a marked step forward in reliability. It is safe to assume that the fuel-injection equipment of the C-I engine will be at least as reliable, if not more so, than the combined ignition and carburetion equipment of the gasoline engine. The simplicity of the mechanism of an oil engine marks it for development toward far greater reliability. An obvious retort to these statements is: Why has it not been done by this time? The answer lies in the relative amount of money which has been spent on the two engine-types; the gasoline engine is highly developed, but the oil engine is just starting in the aircraft field.

No matter how attractive the large oil engine may be to the commercial operator of aircraft, the initial stimulus for its development in the United States probably will come from the military services of the Government. The Army is interested primarily from the standpoint of expected increase in range of its bombers and observation ships, whereas the Navy is not only interested in increasing the range of its patrol boats but seriously desires oil engines primarily from the standpoint of reducing the gasoline fire-hazard on its aircraft carriers, fighting ships, and lighter-than-air craft. The present transport-plane engines are a direct development of the military types, the experimental developments of which, if not paid for in full by the Government, have been substantially encouraged by the military departments, and there is little reason to expect the oil-engine development to take a different turn.

Two-Stroke Versus Four-Stroke Cycle for the C-I Engine.—The C-I engine aspirates and compresses air alone and is therefore inherently suited for two-stroke operation. Further reference to Fig. 3 will reveal the fact that a Diesel cycle must operate with an excess-air figure above 150 to compete with the best Otto-cycle performance. The indicated mean effective pressure is therefore confined to figures less than 120 lb. per sq. in. within the rated-horsepower range. Upward of 25-per cent overload-capacity may be obtained but at the expense of high fuel consumption and smoky exhaust. The rated speed may be assumed at 2000 r.p.m., slightly less than for present large gasoline engines; the rated specific horsepower-output for the four-stroke cycle at 88 per cent mechanical efficiency and the two-stroke cycle at 83 per cent follows as 0.264 and 0.502 hp. per cu. in. Since the four-stroke figure is completely overshadowed by gasoline-engine performance, the two-stroke is the only type worth serious consideration.

Weight Possibilities in Compression-Ignition Engines

Fig. 4 shows the trend of the weight per cubic inch of modern aircraft engines in relation to the size of the individual-cylinder displacement. The lower line represents average data on 25 direct-drive single-row radials, and the upper line follows closely 10 geared radials, both single and two-row. Several points are shown for V-12 engines also. As might be expected, the point for the Jumo-5 engine falls considerably above the trend for in-line engines. This is not primarily a result of the so-called high peak-pressures of the compression-ignition engine but of the design arrangement of the engine. Practically every discussion on the performance of Diesel-type aircraft-engine models justifies the heavy design on the grounds of high peak-pressures and the terrific heat-strains encountered. Such a claim is justifiable in the case of the four-stroke Diesel engines developed to date; for, in order to come within the usable-weight range, highest obtainable mean effective pressures are necessary, consequently resulting in excessive peak pressures and practically Otto-cycle operation, but it does not apply necessarily to the two-stroke engine. Peak pressures of 700 to 800 lb. per sq. in. are encountered in our present spark-ignition high-output engines, and the contemplated higher compression ratios will not lower them. Excellent mean effective pressures have been obtained with top pressures less than 800 lb. per sq. in. on C-I single-cylinder engines at the Massachusetts Institute of Technology and at the engine laboratory of the N.A.C.A. Unpublished results of some commercial experiments are still more encouraging.

It is not generally realized that the top pressure of the true Diesel cycle is not as high as that for the Otto cycle; it is only when the injection characteristics allow the combustion to proceed in the direction of the Otto cycle that excessive peak pressures are encountered. The Lanova system of Diesel combustion for automotive engines, recently introduced in Germany and here, claims to produce peak pressures about equal to normal gasoline-engine pressures as well as corresponding mean effective pressures.

If then, the gas loads of the C-I engine are confined within limits of the present spark-ignition loads, there is little justifi-

cation for C-I designs inherently heavier than the average aircraft engine of similar construction. The two-cycle design might add somewhat to the number of pounds per cubic inch, depending upon the type of porting employed to scavenge the cylinder. Various scavenging means are being developed experimentally over the country, but it is beyond the scope of this paper to touch upon them; let it suffice to say that certain of these scavenging means represent a potential saving rather than an increase of weight when compared to four-stroke valve-gear. Injection equipment will not weigh nearly as much as the conventional carburetion and ignition equipment together. The smoother-output torque-characteristic of two-stroke operation should allow lighter crankshafts. With radial cylinder-arrangement, the two-stroke operation markedly relieves the high connecting-rod bearing-loads encountered with four-stroke engines and thus again permits a saving in basic engine weight.

Now that verbally we have built a C-I engine lighter than a gasoline engine, let us allow for design and fabrication contingencies and place the weight conservatively at 0.75 lb. per cu. in., which will give a specific weight of 1.5 lb. per rated hp., in the power range with large American engines which today fall between 1.4 and 1.5 lb. per hp. Around 850 hp. is the highest rating of any American service engine, and the large planes could easily handle more power. It would seem desirable to take advantage of the weight saving due to scale effect as shown in Fig. 4 and to design for a cylinder size of around 250 cu. in. The Diesel-combustion heat-conditions, not being as severe as in the gasoline engine, would allow a larger cylinder. The heat rejection per horsepower of the Junkers engine is about 70 per cent of the water-cooled engine.

Actual C-I Engine Performance

At this time it is fitting to mention more fully the German Jumo-5 engine. It is the result of the development of the Jumo-4, both of which are two-cycle opposed-piston six-cylinder compression-ignition engines. The design of these engines hinged entirely on obtaining conditions which would produce maximum economy and the best possible scavenging of the cylinder. The engine is worthy of much consideration because it, and its prototypes built by Napier in England and Lilloise of France under Junkers license, are the outstanding successful large C-I aircraft-engines in service. The Jumo-4 is rated at 740 hp. at 1700 r.p.m. and the Jumo-5, a smaller engine, at 532 hp. at 2100 r.p.m. The Jumo design is inherently heavy because constructional weight-considerations were sacrificed. Two crankshafts, each one of which must be designed to carry almost as much torque as the output of both, are placed in separate crankcases about as far apart as is allowable in an aircraft-engine structure and connected together with a train of spur gears. The stroke-bore ratio is made unduly high to obtain the long pipe-like cylinder for good scavenging conditions. The long stroke in the case of the Jumo engines requires not only length of structure for the stroke itself but slightly more again for each piston because the skirt of port-operating pistons must be sufficiently long to cover the ports at top center. In spite of this construction the Jumo-5 weighs but 2.05 lb. per rated hp.; the result of an excellently fabricated design and a high specific output, namely, 0.525 hp. per cu. in.

The American Dechamps engine, an experimental inverted V-12 two-stroke 3050-cu. in. C-I engine, is rated at 1200 hp. at 1600 r.p.m. It weighs 2400 lb. or 0.787 lb. per cu. in. If the speed were raised, this engine would be in the weight-horsepower class with the present designs and away ahead on maximum power.

The horsepower characteristics of the C-I engine versus altitude seem to indicate a distinct advantage over the gasoline engine. Fig. 1 shows the effect of altitude on full-throttle rated-speed horsepower of the Jumo engines compared to a ground-level-rated gasoline-engine. The difference is due to the normal excess air present in the C-I-engine cylinder. Extra altitude-power is obtained at the sacrifice of specific fuel consumption, because the fuel-air ratio rises as the air density decreases. A limit is reached when the C-I engine begins to smoke; from then up the throttle is partly closed and acts as an altitude mixture-control as well as throttle.

Cost of C-I Engines

As with any new product, the introductory price of a competitive oil-engine will be high because extensive development expense will have to be amortized. On a production basis there should be little difference in cost of the basic engine, and what price-difference does exist will result from the accessory costs. The carburetion and ignition equipment for a 700-hp. single-row radial comes close to $600, a figure which should be adequate to cover the oil-injection apparatus of the C-I engine. Provision for compressed-air starting should be designed and built into the engine structure, thus reducing the $400 to $500 inertia-starter tax to a small amount and reducing the installed weight by a substantial figure.

Before marked service operating-economy can be assured, the C-I engine must be capable of digesting diverse grades of commercial fuel-oil. The specially refined Aircraft Diesel oil put on the market to supply the needs of the Packard engine was more expensive than gasoline. The heavier, cheap oils are good Diesel fuels, but relatively poor for C-I engines depending upon Otto-cycle operation to make their ratings.

Specifications for a Future C-I Engine

As a conclusion the following quite plausible specifications are given for a C-I engine which, if it were backed by an effective corporation, would shortly bring the present airline and military engines to earth.

(1) Two-stroke-cycle operation

(2) Adaptability of design up to 1500 b.hp.

(3) Flexibility of cylinder arrangement (small-diameter radial preferred)

(4) Maximum fuel consumption:

	lb. per b.hp-hr.
Clear exhaust, rated b.hp.	0.40
75 per cent of rated b.hp.	0.35
Cruising, 50 per cent of rated b.hp.	0.36
Take-off, 125 per cent of rated b.hp.	0.50

(5) Propeller speed in revolutions per minute arranged with gears for efficient operation.

(6) Must be capable of being supercharged to rated horsepower at 15,000-ft. altitude

(7) Dry weight, 1.5 lb. per rated horsepower, maximum

(8) Peak cylinder-pressure limited to 800 lb. per sq. in.

(9) Air cooled

(10) Starting to be accomplished without glow plugs

(11) Either individual-cylinder injection-pumps or dual unit-pumps

(12) Minimum service-intervals on 75 per cent power operation; 50 hr., routine inspection of nozzles, etc.; 350 hr., top overhaul; and 1000 hr., major overhaul of bearings, pistons, etc.

(13) Engine to have a minimum life of 4000 hr.

(14) Engine must be capable of operating on a good grade of bunker oil

(15) Free of torsional vibration in operating range

The technical data in this paper have been arranged to show the limiting conditions for the comparative efficiencies of the spark-ignition and compression-ignition engine-types, together with a discussion which may help to analyze the efforts toward better fuel economy in aircraft service. It is hoped that our attempts have shed some glimmer of illumination on this very controversial subject of Diesel versus gasoline engines for aircraft. The gasoline engine is so well entrenched and so highly developed that an overnight switching to Diesel power by the flight service seems impossible; still, almost anything can happen in the aircraft industry. The gradual development of aircraft Diesel-power is inevitable and, in time, it may be accepted as the rule instead of the exception; but, in the meantime, the development of the gasoline engine will continue, and give new Diesel developments plenty of competition.

Discussion

Comments on C-I Engine Specifications Stated

—**Leigh M. Griffith**
Consulting Engineer, Los Angeles

COMPARISONS being considered somewhat odious in engineering as well as in other lines, I first want to express my approbation of the attitude of fairness with which Mr. Browne has treated his subject. It is easy to become overenthusiastic on either side of such a controversial subject. My personal interest is in the compression-ignition or Diesel engine, and I feel that the author has presented the case for this engine so well that there is little to add at this time.

The gradual but steady tendency toward airplanes and flying boats of greatly increased size, capacity and speed, is now being handicapped by the lack of correspondingly large engines. The use of more than two engines on the conventional form of large monoplane reduces the effective aspect ratio by increasing the proportion of wing surface subjected to disturbed air flow from the propellers. A fuller appreciation of this condition naturally increases the demand for engines delivering twice or three times the maximum power now available.

Inasmuch as there has been little discussion in the literature regarding the fuel-cost savings made possible by the Diesel engine, the author's figures will be distinctly surprising to many. How long this advantage will be maintained in the face of the constantly increasing use of Diesel fuel is a question of supply and demand, but the Diesel will certainly have an important advantage for many years. The great decrease of fire hazard has been well aired and is so fully appreciated that Diesel-engined aircraft will take the business from gasoline-engined ships, even though the latter may be somewhat faster.

The increased carrying capacity, or decreased take-off weight, made possible by the lower specific-fuel consumption of the Diesel has been discussed at length and the quantitative advantage computed for many specific cases. The fuel rates given by the author are on the whole quite reasonable for the conditions assumed, although there will be vastly more difficulty in getting the gasoline engine to cruise consistently at 0.45 lb., even with the aid of the complicated accessory gadgets and extra-high-octane fuel mentioned, than in maintaining a rate of 0.35 lb. with the Diesel engine. Moreover, in estimating the increased load for a given range, due regard must be paid to the favorable factor of the decreased weight of tanks and tank mountings, made possible by the decreased weight of a more-dense fuel. To complete the picture, the relative lubricating-oil consumptions must also be considered. Although we have few data covering the aircraft-Diesel rate, there is no reason to expect a consumption any higher than for the gasoline engine, except perhaps in the case of the two-stroke Diesel.

Although the higher thermal efficiency of the Diesel will permit us to develop more power from each cylinder for the same unit rate of heat disposal to cooling medium, either by two-stroke operation or by an enlarged cylinder operating four-stroke, it certainly appears desirable to adopt liquid cooling for the 1500-hp. and larger units now needed. Furthermore, the liquid-cooled engine can be mounted inside the wing on large monoplanes, where it can readily be given the benefits of continuous attendance in flight and many of the petty failures nipped in the bud, to say nothing of the increased aerodynamic efficiency obtained by such enclosure.

I certainly do not agree with the author's statement that the Diesel "must operate with an excess-air figure above 150 to compete with the best Otto-cycle performance." The modern high-speed Diesel of the open-combustion-chamber type will show a high efficiency with as little as 15 per cent excess air. In fact, his curve for the Jumo-5 shown in Fig. 3 should be moved to the right, the higher point shown being for approximately 20 per cent excess air, instead of for 60 per cent as shown. The brake efficiency then checks well with Fig. 2.

The author's "Specifications for a Future C-I Engine" are well drawn and I agree with most of them. As to (2), I feel that 1500 hp. should be the starting point, rather than the ultimate objective. The most important application at present involves the replacing of four 750-hp. engines by two 1500-hp. Diesels. The small-diameter radial recommended in (3) is not well adapted to liquid cooling, and the air-cooled type cannot be placed within the wing. I strongly disagree with (9), for reasons previously stated. In the matter of the fuel-injection system (11), I am in favor of a modified common-rail system as being on the whole simpler, more reliable and easier to maintain, as well as making possible that definite and accurate control of fuel injection necessary to produce the combustion characteristics desired. Except possibly for Naval use, I see no reason to try to burn bunker oil (14).

For reasons partially stated by the author, these large Diesels will be two-stroke. In order to develop 1500 hp., such a two-stroke engine will have say 16 pistons of 6.00-in. diameter and 8.00-in. stroke. High mechanical efficiency and low engine weight being of great importance, and the efficient scavenging offered by the opposed-piston design being considered necessary, the best arrangement appears to be the opposed "barrel" design embodying low-friction swash-plate mechanisms. Because of the elimination of piston side-load, the transmission of piston thrust through non-sliding bearings, the complete balance obtained, the use of a short and rigid plain shaft of large diameter, the very complete elimination of material not continuously worked, the use of ball or roller bearings in the swash plates and main bearings, the compact arrangement and the accessibility, the barrel-cylinder arrangement should be subjected to further intensive development to meet this high-power Diesel need.

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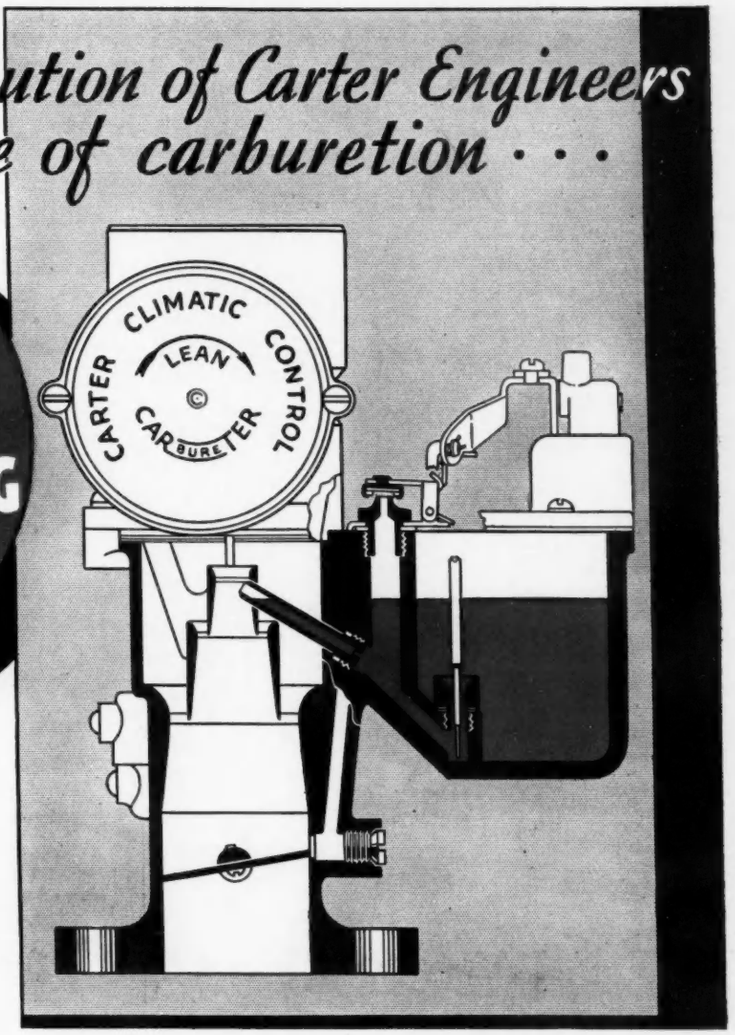
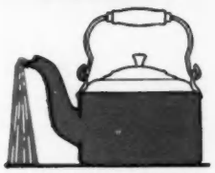
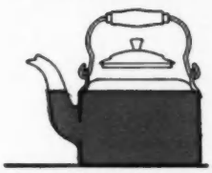




PERCOLATING



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Notes and Reviews

THESE items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a rule no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.

AIRCRAFT

Tank Tests of Model 11-G Flying-Boat Hull

By J. B. Parkinson. N.A.C.A. Technical Note No. 531, June, 1935; 7 pp., 1 table, 14 figs. [A-1]

Method of Testing Oxygen Regulators

By Harcourt Sontag and E. L. Borlik. N.A.C.A. Technical Note No. 532, June, 1935; 7 pp., 2 figs. [A-1]

Spinning Characteristics of Wings—I-Rectangular Clark Y Monoplane Wing

By M. J. Bamber and C. H. Zimmerman. N.A.C.A. Report No. 519, 1935; 20 pp., illustrated. Price, 10 cents. [A-1]

The Influence of Wing Setting on the Wing Load and Rotor Speed of a PCA-2 Autogiro as Determined in Flight

By John B. Wheatley. N.A.C.A. Report No. 523, 1935; 4 pp. with figures. Price, 5 cents. [A-1]

Air Flow in a Separating Laminar Boundary Layer

By G. B. Schubauer. N.A.C.A. Report No. 527, 1935; 12 pp., illustrated. Price, 5 cents. [A-1]

An Analysis of Longitudinal Stability in Power-Off Flight With Charts for Use in Design

By Charles H. Zimmerman. N.A.C.A. Report No. 521, 1935; 31 pp., with tables and charts. Price, 10 cents. [A-1]

Ground Effect on the Take-Off and Landing of Airplanes

By Maurice Le Sueur. Translated from *La Science Aérienne*, Vol. III, No. 1, January-February, 1934. N.A.C.A. Technical Memorandum No. 771, July, 1935; 32 pp., 23 figs. [A-1]

The Denis-Gruson Six-Component Wind-Tunnel Balance

N.A.C.A. Technical Memorandum No. 772, July, 1935; 6 pp., 6 figs. [A-1]

Two Notes on the Boundary Influence of Wind Tunnels of Circular Cross Section

By Itirô Tani and Masuo Taima. Report No. 121 of the Aeronautical Research Institute, Tokyo Imperial University; June, 1935; 27 pp., illustrated. [A-1]

Spar Design

By J. F. Cuss. Published in *The Aircraft Engineer*, supplement to *Flight*, June 20, 1935, p. 41. [A-1]

A simplified method of determining the sizes of flanges in spruce spars for monoplane wings is set forth. The author explains that the design procedure discussed is particularly applicable to the main plane spars of a cantilever monoplane, but the principle may be applied equally well to any other beam of similar construction.

The Short "Sarafand"

Published in *Flight*, June 13, 1935, p. 631. [A-3]

This is the first published description of the largest British flying boat which has a loaded weight of 70,000 lb. and a speed of 150 m.p.h. The "Sarafand", known at first as the Short R.6/28, was primarily an experiment designed to ascertain the possibilities of the type. A civil

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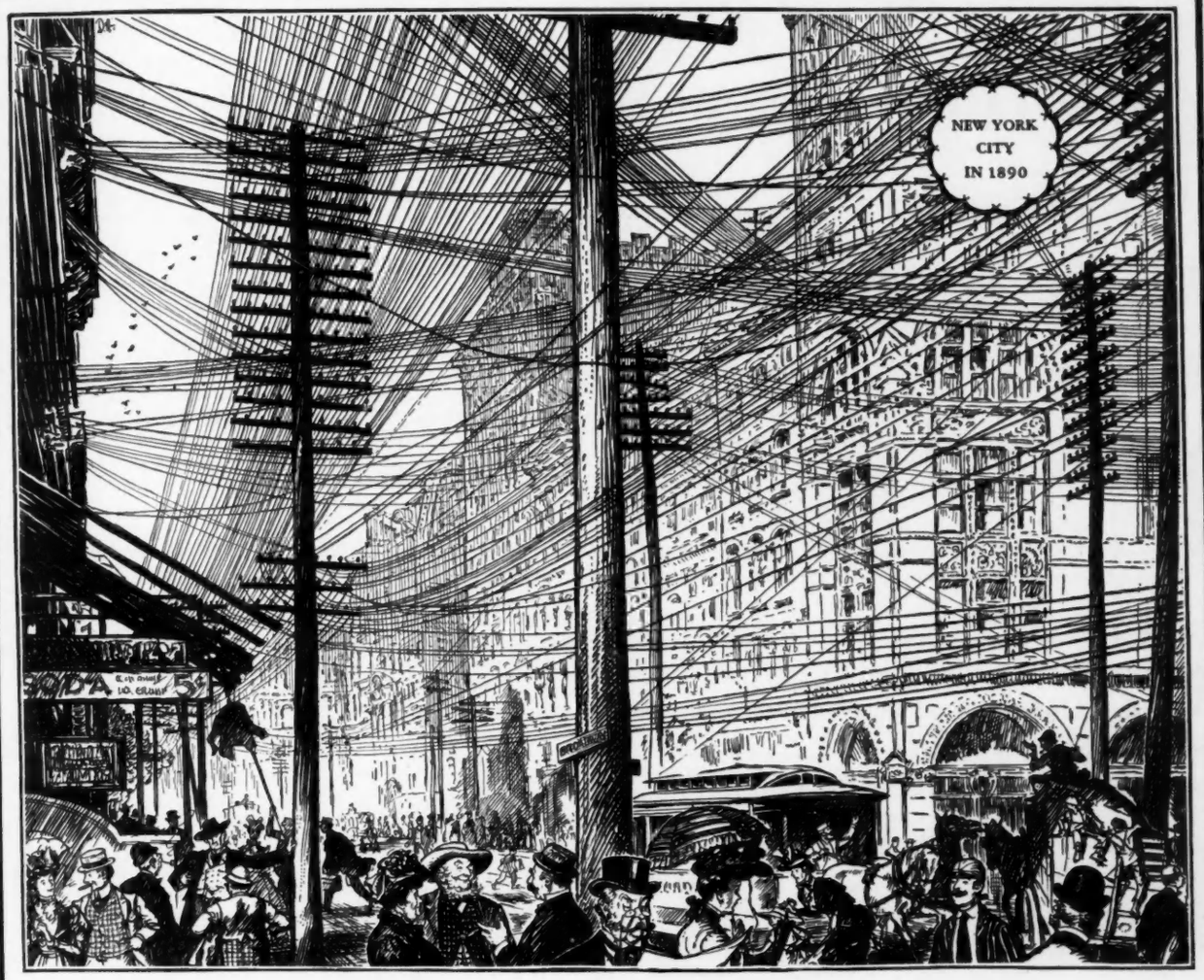

Years of Progress

HAVE BROUGHT MANY IMPROVEMENTS IN TELEPHONE SERVICE

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This has meant a series of conquests of space, and insured greater clarity and dependability for every telephone user. But it is only one of many kinds of improvements that have been made.

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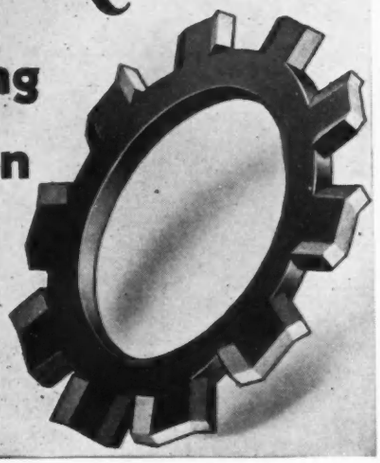

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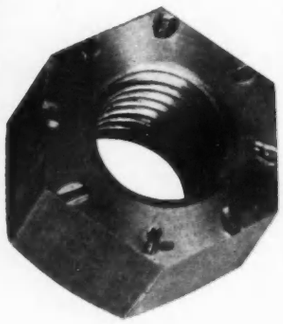

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NOTES AND REVIEWS

Concluded

machine of similar dimensions was projected about the same time, but in the interests of economy its completion was abandoned.

In its general conception the Short "Sarafand" is a normal biplane flying boat of all-metal construction, the most notable features of the general lay-out being the thickening of the lower wing-roots to avoid the necessity for chine struts, and the placing of the six engines in three tandem pairs, each pair being carried on a single pair of inter-plane struts.

The Aircraft Year Book for 1935

Compiled and published by the Aeronautical Chamber of Commerce of America, inc., New York City, 1935; 528 pp., illustrated. [A-3]

The "official and complete record of American aviation" for the past year is now available.

The Third Deutsch Cup

By R. J. de Marolles. Published in *Aircraft Engineering*, June, 1935, p. 139. [A-3]

The author describes the race with full details of the Caudron Entrants and Renault Engine.

Instrument and Radio Flying

By E. A. Cutrell. Published in *Aviation*, June, 1935, p. 11. [A-4]

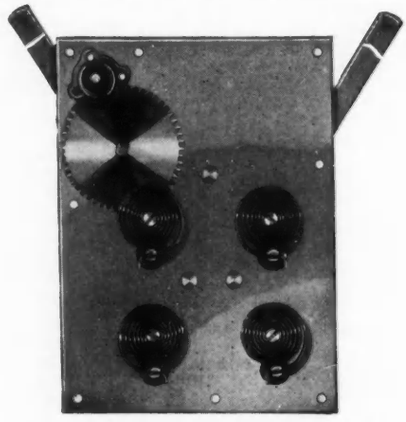
Out of his long experience as a member of the Department of Commerce piloting staff engaged in blind landing research, as instructor in instrument flying for American Airlines, the author outlines certain practices which he has found useful in instrument flight training and radio beam flying, and discusses some of the faults and difficulties which are commonly encountered.

BODY

Product Appeal Through Correct Color

By Howard Ketcham. Published in *Product Engineering*, July, 1935, p. 257. [B-3]

This is the first in a series of articles in which Mr. Ketcham, Editor of the *Automobile Color Index*, gives convincing reasons why color plays a strong hand in attracting or deflecting business.



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